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OF

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The professional papers contained in The Journal are published prior to the meetings at which they are to be presented, in order to afford members an opportunity to prepare any discussion which they may wish to present.

The Society as a body is not responsible for the statements of facts of opinions advanced in papers or discussions. C55

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VOL. 32 OCTOBER 1910 NUMBER 10

COMING MEETINGS

The first meeting of the Society for the coming season will be held in the Engineering Societies' Building, New York, on Tuesday evening, October 11. The paper will be by Frank B. Gilbreth; Mem.Am.Soc.M.E., on Fires: Effects on Building Material and Permanent Elimination. This paper was published in The Journal for May and lantern slides will be used showing the results of the author's observations upon the effect of fires in the great conflagrations at Baltimore, San Francisco, Chelsea and elsewhere. Following this, H. deB. Parsons, Mem.Am.Soc.M.E., will show lantern slides illustrating the effect of fire on tall buildings. Prof. Ira H. Woolson, Mem. Am.Soc.M.E., Henry M. Keasby of the National Fire Proofing Co., Wm. D. Grier, Chairman of the National Fire Protection Association, and others concerned with the use of concrete and other fire-proof materials and with fire protection will participate in the discussion.

MEETINGS IN BOSTON

Meetings of the Society in Boston are announced for October 19, November 11 and December 23. The October meeting will be conducted by the Boston Society of Civil Engineers, The American Society of Mechanical Engineers and other engineering societies of the city coöperating. The subject will be An Account of the Destruction of Carthego, Costa Rica by the Earthquake of May

4, 1910. Prof. Thomas A. Jaggar will speak on the geological features and Prof. Chas. M. Spoffard will discuss the effects of the earthquake on different classes of structures. Views will be shown on Carthego taken soon after the earthquake and by photographs of Central American scenes.

The November meeting will be conducted by The American Society of Mechanical Engineers and the December meeting by the Boston Section of the American Institute of Electrical Engineers. It is expected that the meeting in January will take the form of a dinner of the character of the one held last January at which there were some five hundred in attendance, representing the various engineering and architectural societies of Boston and vicinity and many of the technical schools and other organizations.

ANNUAL MEETING

The Annual Meeting of the Society will be held in New York, December 6-9, 1910, the assignment of Papers for the meeting will be made from those published in this number of The Journal and in the November and December Journals. The program for the meeting is in preparation and will be announced as early as possible.

RAILROAD TRANSPORTATION NOTICE

Arrangements for hotel, transportation and Pullman car accommodations should be made personally.

For members and guests attending the Annual Meeting in New York, December 6-9, 1910, the special rate of a fare and three-fifths for the round trip, on the certificate plan, is granted when the regular fare is 75 cents and upwards, from territory specified below.

a Buy your ticket at full fare for the going journey, between December 2 and 8 inclusive, and get a certificate, *not a receipt*, securing these at least half an hour before the departure of the train.

b Certificates are not kept at all stations. If your station agent has not certificates and through tickets, he will tell you the nearest station where they can be obtained. Buy a local ticket to that point and there get your certificate and through ticket.

c On arrival at the meeting, present your certificate to S. Edgar Whitaker, office manager, at the Headquarters. A fee of 25 cents will be collected for each certificate validated. No certificate can be validated after December 9.

d An agent of the Trunk Line Association will validate certificates, Dec. 7, 8, 9. No refund of fare will be made on account of failure to have certificate validated.

e One-hundred certificates and round trip tickets must be presented for validation before the plan is operative. This makes it important to show the return portion of your round trip ticket at Headquarters.

f If certificate is validated, a return ticket to destination can be purchased, up to Dec. 13, on the same route over which the purchaser came, at three-fifths the rate.

This special rate is granted only for the following:

Trunk Line Association:

All of New York east of a line running from Buffalo to Salamanca, all of Pennsylvania east of the Ohio River, all of New Jersey, Delaware and Maryland; also that portion of West Virginia and Virginia north of a line running through Huntington, Charleston, White Sulphur Springs, Charlottesville, and Washington, D. C.

Central Passenger Association:

The portion of Illinois south of a line from Chicago through Peoria to Keokuk and east of the Mississippi River, the States of Indiana, and Ohio, the portion of Pennsylvania and New York north and west of the Ohio River, Salamanca and Buffalo, and that portion of Michigan between Lakes Michigan and Huron.

Eastern Canadian Passenger Association:

Canadian territory east of and including Port Arthur, Sault Ste. Marie Sarnia and Windsor, Ont.

GENERAL NOTES

NATIONAL CONSERVATION CONGRESS

The second National Conservation Congress, held in St. Paul, Minn., September 5–9, 1910, called forth much discussion on the subject of the control of national resources. President William H. Taft, who opened the congress, sounded the key-note of its policy of the conservation of these resources under national regulation and control, discussing the question under seven heads, namely agricultural, mineral, forest, coal, oil and gas, and phosphate lands, and water power sites. Ex-President Theodore Roosevelt in his speech on the following day emphatically urged, in addition to Mr. Taft's recommendations, that the federal government retain control of water power sites as well as coal lands and forests. Other speakers, among them James J. Hill, urged that these resources be left to the control of the individual states, believing the national machine to be too large and too remote and its operation too slow, cumbrous and costly.

The Congress concluded its deliberations by adopting a platform in which it declared the rights of the people "to be natural and inherent and justly inalienable and indefeasible" and that these resources must be "developed, used and conserved in ways consistent both with current welfare and the perpetuity of our people." It further declared that the waterways should be administered in the interests of all the people, and that in doubtful cases there should be coöperation between state and federal agencies; that use of the waters should be made with reference to all other uses and care be taken to prevent defilement; that the policy with regard to swamp lands be continued and extended; that a federal commission empowered to deal with all water powers be maintained; that public lands be withdrawn pending classification and surface rights be separated from mineral, forest and water rights; that mineral lands be leased for a limited period; and that national forests be controlled by the federal government and the forestry service be given larger appropriations. The Congress further expressed itself as opposed to child labor as detrimental to the life and development of the people physically, intellectually and morally, and recommended legislation suitable to this end.

The Society was represented at the Congress by Paul Doty, who acted as chairman of the local board of managers; by E. E. Johnson, J. J. Flather, L. H. Gardner and M. E. R. Toltz.

AMERICAN INSTITUTE OF MINING ENGINEERS

Preparations for the meeting of the American Institute of Mining Engineers, to be held in November in the Canal Zone, are proceeding satisfactorily and more than 100 members and guests have signified their intention of attending. The party will sail from New York on October 21, on the steamer Prinz August Wilhelm of the Hamburg-American line, and the technical sessions will probably be held en-route. Arrangements have been made for the accomodation of the visitors during their stay at the Hotel Tivoli in the Canal Zone.

According to the program now being arranged, the party will arrive at Colon on November 1 and proceed by train to the Gatun Dam, where after luncheon an inspection will be made of the dam and locks. On November 2 a trip will be made by train through the Culebra cut. November 3 is the anniversary of the independence of the Republic of Panama and is observed as a public holiday, with a popular celebration in the city of Panama, including masquerades, fire-works, buli-fights, etc., and no other plans will be made for that date. On November 4 there will be an excursion to the locks and dams at Pedro Miguel and Miraflores; on November 5 an excursion by tug to the Pacific entrance of the Canal, the islands in Panama Bay, etc.; and on November 7 an excursion by train over the relocated line of the Panama Railroad, now in course of construction. The following day will be occupied in the return to the steamer at Colon.

It is further planned to stop en route at Havana and Kingston, and returning at Port Limon, Kingston and Fortune Island, arriving in New York on November 15.

The rapid progress of work on the Isthmus which will in the course of not very many months obscure the many features of construction and methods of operation now open to inspection, makes a trip to Panama at this time especially valuable and interesting to engineers.

The following members of the Society have accepted the invitation of the American Institute of Mining Engineers to join in this meeting: Prof. William Kent, Charles Kirchhoff, R. V. Norris, H. H. Porter, David B. Rushmore, C. M. Russell, W. L. Saunders, F. W. Scarborough, John M. Sherrerd, S. D. Warriner and Walter Wood.

DR. J. A. HOLMES DIRECTOR OF THE NEW BUREAU OF MINES

Dr. Joseph Austin Holmes, Mem.Am.Soc.M.E., who has just been appointed director of the new Bureau of Mines, has for some time past been an eminent officer of the Geological Survey. Mr. Holmes was graduated from Cornell in 1881, and after visiting extensively the mining regions of this country and Europe, occupied for ten years the chair of geology in the University of North Carolina. He resigned from this work to become state geologist of North Carolina and was appointed to the head of the department of mines and metallurgy at the St. Louis Fair in 1904. When the Government fuel investigations were authorized by Congress Dr. Holmes was offered its supervision and has since then been engaged in the examination of coal-fields in a way which has aided both consumers and producers. Dr. Holmes has done much to introduce safety devices into the mines and inaugurated the recent movement for mine rescue work, which includes three state and twelve government mine rescue stations.

AMENDMENT TO BY-LAW 22

The following is the correct text of the amendment to By-Law 22:

B 22 The Finance Committee shall consist of five Members or Associates. The term of office of one Member of the Committee shall expire at the end of each Annual Meeting. This Committee shall, under the direction of the Council, have a supervision of the financial affairs of the society, including the books of account. The Committee may cause the accounts of the Society to be audited and approved annually by a chartered or other competent public accountant. The Committee shall hold monthly meetings for the audit of bills and such other business as shall come before it and shall deliver to the Secretary for presentation to the Council at the end of each fiscal year, a report of the financial condition of the Society for the past year, and also shall present therewith a detailed estimate for the probable income and expenditure of the Society for the following twelve months. It shall make recommendations to the Council as to investments, and when called upon by the Council, advise upon financial questions. It shall have charge of the making of all contracts and other obligations to pay money in the Society's work and the ordering of all expenditures thereunder.

NECROLOGY

CHARLES T. PORTER

Charles Talbot Porter, Hon.Mem.Am.Soc.M.E., a charter member of the Society and recipient of the John Fritz Medal for his "work in advancing the knowledge of steam engineering and for improvements in engine construction," died in New York, August 28, 1910, at the home of his son, Louis Morgan Porter.

Mr. Porter was born at Auburn, N. Y., January 18, 1826. He was descended from a notable line of New England ancestors including, on his father's side, the Rev. Jonathan Edwards, and on his mother's side, Governor John Winthrop of Massachusetts and Governors Saltonstall and Winthrop of Connecticut. He was graduated from Hamilton College in 1845 and at the fiftieth reunion of the class presented the Half-Century Annalist's Letter, a feature of the annual meetings of the Hamilton alumni. After graduation he read law in his father's office in Auburn, and was admitted to the bar in1847. The next year he married Harriette S. Morgan of Aurora, N. Y., and their married life was unbroken until within a few weeks of Mr. Porter's death, when Mrs. Porter died.

After practicing his profession for six or seven years, first at Rochester and afterwards in New York City, Mr. Porter became interested in mechanics in connection with a stone-dressing machine invented by one of his clients, which failed to operate satisfactorily. Believing that the fundamental principles of the machine were correct Mr. Porter went to work to improve it, picking up by the way a knowledge of drafting and designing, and therein brought out his latent mechanical ability. The stone-dressing machine was driven by a steam engine which he desired to run at high speed, but the governor was of the usual simple fly-ball type which could not be speeded up and consequently the regulation of the engine was faulty. To remedy this defect Mr. Porter was led to design and perfect his well-known central counterpoise type of governor which has since carried his name.

Subsequently came the development of the high-speed Allen steam engine, later known as the Porter-Allen engine, which was essentially

the life-work of Mr. Porter. The first engine was built in this country and shown at the London Exhibition of 1862, equipped with the Porter governor, and operating non-condensing. In 1867 at the French Exposition five engines were installed, the only high-speed engines exhibited.

The exhibit at London and subsequent attempts to sell engines of this type in England showed the demand to be entirely for condensing engines. This brought about the development by Mr. Porter of a jet condenser to be direct-connected to his engines, with an air pump adapted to the high speed at which the engines ran. The building of these engines was begun in England in 1864.

In his work in steam engineering Mr. Porter became associated or intimately acquainted with many of the early distinguished engineers, notably with John F. Allen and Charles B. Richards. Mr. Allen had originated a link and valve motion for steam engines, well adapted for use with the Porter counterpoise governor, and it was the combination by Mr. Porter of this mechanism with his governor, together with Mr. Porter's advanced ideas upon high rotative speeds and methods of engine construction that resulted in the Porter-Allen engine.

Along with the study of steam economy was the need of a steam-engine indicator adapted to high speeds. This led to the design by Mr. Richards of the first indicator to meet these requirements. The patents were acquired by Mr. Porter and an instrument was shown in connection with the engine at the London Exhibition. It was shortly afterwards manufactured by Elliott Brothers of London.

In the early manufacture of his engines, many practical difficulties had to be met owing to the crudeness of machine shop methods. Numerous devices and systems of manufacturer were introduced by Mr. Porter to attain the accuracy, without which successful high-speed machinery would be impossible.

In 1868 Mr. Porter returned from England, formed a partnership with Mr. Allen and began the manufacture of engines in a small shop in Harlem, N. Y. During the three years of business depression, beginning with 1873, the manufacture of the engines was discontinued, but later was begun at the Hewes and Phillips Iron Works at Newark, N. J., under Mr. Porter's own name. They have since been manufactured by the Southwark Foundry and Machine Company at Philadelphia.

In 1880 Mr. Porter installed a high-speed steam engine in the Edison laboratory at Menlo Park, N. J., which marked the beginning of direct-connected generators. Following this, the first of a series of

engines for so-called steam dynamos was constructed for the Edison Station at Pearl Street, New York, each independently driven by a direct-coupled engine.

While these events are important in the history of the high-speed engine for electric generating, the introduction of Mr. Porter's engines into rolling-mill work was of even greater moment. The early processes were deliberate because man was habituated to slow movements. The first power came from the slow-turning water wheel, later from the slow-speed steam engine. Faster movements were obtained through gears and belts and then came the direct-connected, easily controlled high-speed engine.

At the first Annual Meeting, in 1880, of The American Society of Mechanical Engineers, Mr. Porter read a brief paper upon The Strength of Machine Tools, and he has since presented numerous others. At the beginning of the manufacture of the Richards indicator, he prepared for the makers, Elliott Brothers of London, a brief treatise on the Steam Engine Indicator, and in 1874 this was revised and very much enlarged by him and brought out simultaneously in London and New York. This contained the tables of the properties of saturated steam which so long remained a standard, based upon the experiments of M. Regnault. Very recently he published his Engineering Reminiscences, which are an interesting and valuable account of many incidents in the development of steam engineering. Mr. Porter was a member of the Board of Judges at the Centennial Exposition of 1876.

ALBERT SPIES

On August 16, 1910, Albert Spies, a Member of the Society, editor and proprietor of Foundry News, died suddenly at his home 40 Glenwood Ave., Jersey City, N. J. He was born in New York City, July 20, 1862, and received his early education from tutors and at the Hoboken Academy. When only 19 years old he graduated from Stevens Institute of Technology, and for a number of years engaged in engineering practice and technical journalism. In 1893 he became editor of Cassier's Magazine, and in 1904 also assumed the editorial conduct of the Electrical Age. For several months he brought out both papers practically unaided until the pressure of work forced him to relinquish his activities on the latter journal in January 1906. In the fall of 1906, following the death of Louis Cassier, Mr. Spies severed his connection with both papers and in February 1907 became

editor of the Electrical Record, continuing until early in this year when he resigned to bring out his own paper, Foundry News. Only four issues were published prior to his death.

Mr. Spies was a member of the Engineers' Club and the American Institute of Mining Engineers, as well as an Associate member of the American Institute of Electrical Engineers. In 1895 he married Miss Gretchen A. Weisker, who, with a son, survives him.

JOHN EDWARDS McKAY

John Edwards McKay was born February 22, 1837 in New York, and received his early education in the public schools and in the Free Academy, now the College of the City of New York. In 1852 he began a two years apprenticeship with Charles W. Copeland of New York and in 1854 entered the drawing-room of the Morgan Iron Works as draftsman and machinist. In May 1859 he was appointed third assistant engineer, U.S.N., from which post he resigned after serving one year, and was again employed by Copeland and by the Coast Survey. In August 1862 he was reappointed to his former grade in the Navy, in December promoted to second assistant, and in May 1864 made first assistant, retaining this position for two years. At that time men in the service of the Navy with practical experience as well as ability to design and make drawings were much in demand, and as Mr. McKay was well equipped in these respects he was ordered to shore duty at Washington, D. C., where under the direction of Benjamin F. Isherwood, Hon.Mem.Am.Soc.M.E., chief of the Bureau of Steam Engineering, he made designs and drawings of marine engines, boilers, etc., for many of the government war-boats. From 1868 to 1872 Mr. McKay was superintendent of the Wood and Mann Engine Company, Utica, N. Y., designing and constructing many steam engines, rolling-mills, saw-mills, etc. He invented at this time a very excellent automatic variable cutting-off for steam engines.

In 1872, at the request of Mr. Tracey, chief engineer of the Croton Aqueduct, Mr. McKay severed his connection with the Wood and Mann Company and entered upon a public service career. During the long period dating from 1872 until his death on May 12, 1910, Mr. McKay was connected successively with the Department of Public Works, the Department of Water Supply and the Department of Water Supply, Gas and Electricity of New York City, and was prominently identified with the many important improvements and

extensions in connection with the water supply of the Boroughs of Manhattan and the Bronx, as they are now called, and with the care and maintenance of the system.

Included in the work done by Mr. McKay was the design and supervision of the construction of the tower, chambers, gates, screens, pipes, etc., in connection with the storage reservoir on the west branch of the Croton River at Boyds Corners, Putnam County, N. Y.; the design and preparation of the plans for the storage reservoir on the middle branch of the Croton River near Brewster's Station, Putnam County, N. Y.; the design and drawing of the junction gate-houses and appurtenances, at 113th Street and Amsterdam Avenue and at 93d Street near Columbus Avenue, New York City, and the drawings of the six lines of four-foot pipes between these gate-houses for the replacement of the old aqueduct between those points; the design and drawings of the first successful submarine pipe between Manhattan and Blackwells Islands for a supply of Croton water to the institutions on the latter; in 1876 the design and drawings and superintendence of the installation of the five-million gallon pumping engine now in the High Bridge Pumping Station, and also the superintendence of the six-million gallon pumping engine in that station; the original studies and plans of the High Service Pumping Station, machinery, boilers, stand-pipe and tank, etc., between 97th and 98th streets near Columbus Avenue, New York City; the design and drawings and the superintendence of installation of a specially designed quick-action 36-inch valve in the gate-house of the Central Park reservoir for the control of a fir line to the lower part of Manhattan; drawings in connection with the Fourth Avenue improvement work from 42d Street to the Harlem River and the supervision of construction on this work. From 1903 until the time of his death Mr. McKay had responsible charge of the Croton, the Bronx and the Byram water sheds and the storage reservoirs there, including the dams, gate-houses, aqueducts, pipelines and appurtenances and the distributing reservoirs in the city, and was responsible for the proper conservation of the storage and the flow through the aqueducts to New York City.

Besides being a member of this Society, Mr. McKay was a member of the American Society of Civil Engineers and the Engineers Club. At the time of his death Mr. McKay resided in White Plains, N. Y.

CHARLES B CLARK

Charles B. Clark, Member of the Society, was born at Bangor, Me., June 20, 1858, and died March 24, 1910. He was educated in the public schools of Bangor and in 1876 began an apprenticeship of five years with the Hardy Machine Company, Biddeford, Me., during which time he studied mechanical engineering under a private tutor. In 1881 he was employed as an engineer on a steamboat and in 1882 worked as journeyman in the shops of the Lawrence Machine Company, Lawrence, Mass., where he also attended the evening technical schools. He was subsequently employed as superintendent of the construction of machinery for the Detroit Dry Dock, Detroit, Mich., the Dunford and Alverson Dry Dock, Port Huron, Mich., and the Simpson Dry Dock, East Boston, Mass. In 1889 he returned to the Lawrence Machine Company as superintendent and general manager, leaving this concern three years later to become manager of the Orono Pulp Company, Orono, Me. Four years later he went to New York where from 1896 to 1898 he acted as manager of the Merrimac Paper Company of Lawrence, Mass. Since February 1898 he had been general superintendent of the Eastern Manufacturing Company's mills, South Brewer, Me.

CHARLES A. FERRY

Charles A. Ferry, Associate Member of the Society, died at Phoenix, Arizona, May 2, 1910. He was born at Utica, N. Y., September 1, 1851 and prepared for college at the Montreal High School. After a year at McGill College he entered Yale from which he was graduated in 1872 with the degree of B.A. He remained in New Haven for three years doing graduate work and in 1875 went to Chicago where he studied law in the office of Judge Upton. Shortly afterwards he became interested in manufacturing and was engaged in the manufacture of steel railway couplers, in 1891 becoming president of the Chicago Tire and Spring Company. For several years before his death Mr. Ferry was not engaged in active business, but was always interested in the development of the science of modern times as applied in all directions.

JOSEPH GARBETT

Joseph Garbett, Member of the Society, who was for many years prominently identified with the mechanical industry of Minneapolis, died at his home in that city August 22, 1910. He was born in Shropshire, England, January 7, 1852, in which town he received his early education, and where he spent the first years of his commercial life in the Horshay Iron Works, and later was associated with the Coldbrookdale Iron Works, becoming superintendent of the latter at the age of twenty-three. In 1879 he left England, going direct to Minneapolis and entering the employ of O. A. Pray and Company, manufacturers of flour and saw-mill machinery, as foreman and patternmaker. He steadily developed into a thorough and able mechanic and when the Twin City Iron Works was established he, with O. P. Briggs, constituted the firm, Mr. Garbett serving as chief engineer. It was here that he designed the Twin City Corliss engine which is so favorably known throughout the country today. When the Twin City Iron Works was absorbed by the Minneapolis Steel Machinery Company in 1902, Mr. Garbett was prominent in its affairs, becoming mechanical engineer of that company as well as a director.

In steam engineering Mr. Garbett was an accepted authority. For the past five years he had devoted much time to gas engines and gas producers. He spent several years abroad studying the various types of foreign power plants of this type, and finally adopted the Muenzel, a German engine, as the foundation of a gas engine for American manufacture. To this he applied his skill and engineering ability and developed one of the most successful gas engines and suction producers in use in this country today.

THE ROTARY KILN

By Ellis Soper, Detroit, Mich.

Member of the Society

Early Development. In the manufacture of Portland cement, the methods used at the present time are quite crude, but even so, the progress made in many departments has been very rapid, particularly in the burning department. In approximately 90 per cent of the mills in this country, pulverized coal is used as a fuel and with few exceptions the total burning cost, including the fuel, represents from one-third to one-half of the total cost of manufacture per barrel. Improvements on the present system are daily being made and experiments on a large scale are being carried on by many of the manufacturers.

2 It must be remembered that the first rotary kiln was manufactured about 1885 and not until 1895 was the rotary type considered a success. The plants of Germany and England and also of this country utilized vertical or stationary kilns, which are much more economical in point of fuel consumption but very costly on account of the excessive hand labor necessary.

3 In 1885 Ernest Ransom patented a rotary kiln in England. About a year later Alphonse de Navarro purchased the American rights to this patent and built the first rotary kiln in this country in a mill from which the present great Atlas Company was evolved. It was 5 ft. in diameter by 40 ft. long. The first fuel tried was wood, but a sufficient temperature for clinkering could not be obtained and petroleum was utilized. The cost of the petroleum became excessive and in 1895 (only 15 years ago) pulverized coal was first tried as a fuel in the Atlas mill. To the American engineer and financier is due the wonderful growth of the business and especially the development of the rotary kiln. It was used commercially and developed here before being adopted in Germany and England where the industry originated and was much older. The development of the kiln from

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

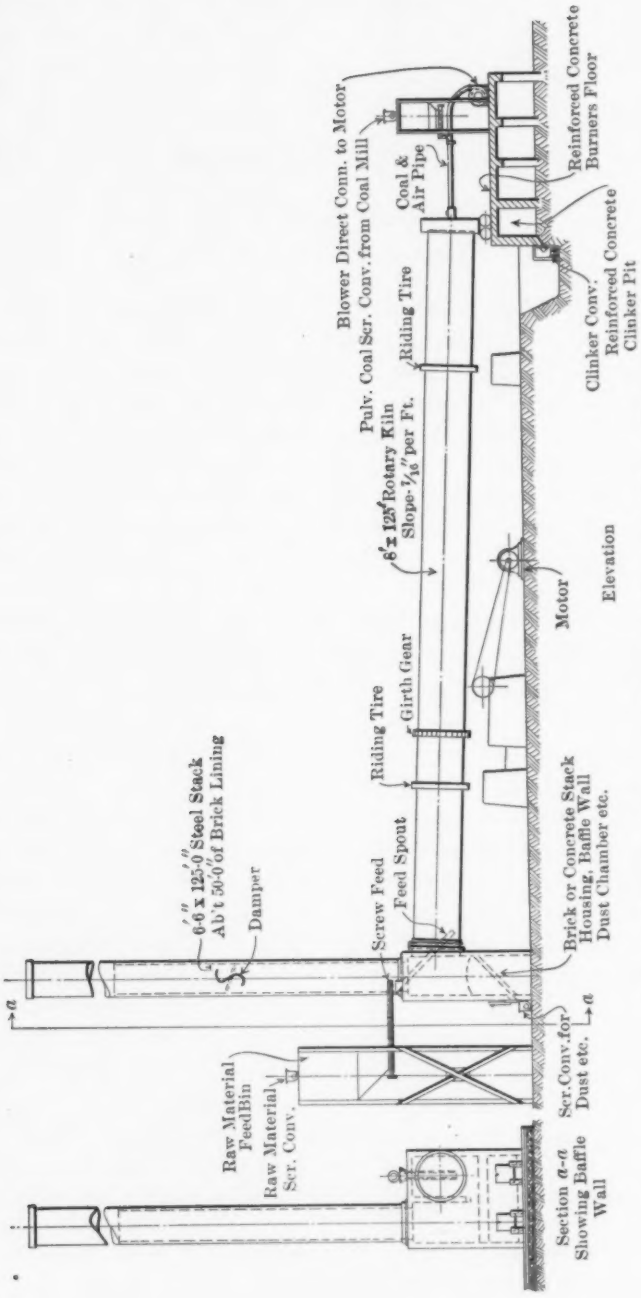


FIG. 1 TYPICAL INSTALLATION OF AN 8-FT. BY 125-FT. ROTARY KILN

its original size of 5 ft. in diameter by 40 ft. long to one 12 ft. in diameter by 232 ft. long indicates a remarkable growth. Whether the limit in size has been reached is a question yet to be determined. Until the last three or four years the 6-ft. by 60-ft. kiln was the standard, and when Mr. Edison installed his 9-ft. by 150-ft. kilns he was laughed at, but the present manufacturers have him to thank for the biggest single advance step in the history of the industry. The main ideas in developing the rotary kiln appear to be increase in output, decrease in fuel consumption per barrel and decrease in the amount of machinery operating; in other words, *concentration*. But it is questionable which is the better proposition, a mill with one large unit producing 2000 barrels per day, or a mill with four smaller units producing the same output. Allowing for the ordinary operating difficulties and shutdowns due to repairs and other ordinary causes, the total output of the one-unit mill, we believe, will be considerably less than that of the four-unit mill. Whether the saving in fuel consumption of the larger unit will make up for this decrease in production is a question to be considered.

4 *Uses of the Rotary Kiln.* The rotary kiln has lately been utilized successfully in burning lime, drying materials of varied character and in driving off the oxides in iron ores. A 7-ft. by 100-ft. kiln is being successfully used in burning lime, and a production of 9 lb. of lime to 1 lb. of fuel has been secured. The process is continuous and we believe it will eventually be widely adopted. As a direct-heat rotary dryer, it is the best in point of production that can be installed and where a large production is desired, a proper installation is ultimately as economical as the majority of the so-called patent dryers.

5 The Jones "Step-Process" of reducing iron ore direct to metallic iron by use of the rotary kiln is very interesting, and is now practically past the experimental stage. Crushed ore is passed through a rotary kiln 8 ft. by 120 ft., in which a reducing flame is maintained. The coal saving over the present blast furnace method is approximately 70 per cent. The following is a comparative statement of the fuel consumption per ton of steel:

Jones Step Process		Ordinary Process	
Kiln for heat	225 lb.	Blast Furnace	3000 lb.
Kiln for reduction	400 lb.	Puddling Furnace	500 lb.
Puddling Furnace	500 lb.		
Total	1125 lb.	Total	3500 lb.

Saving, 2375 lb. fuel per ton steel

6 *The Rotary Kiln in Cement Manufacture.* The most important use of the rotary kiln, however, is in the burning of cement clinker. The following is a general description of the design, installation, operation, etc., of an 8-ft. by 125-ft. kiln, together with some tests on kilns of other sizes.

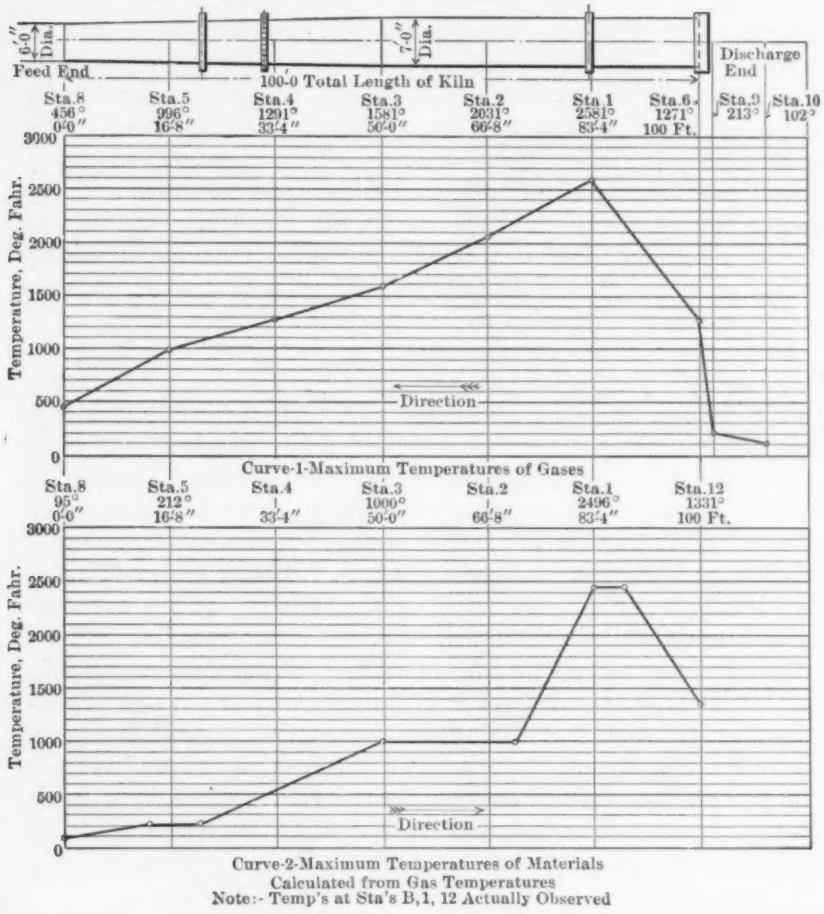


FIG. 2 TEMPERATURE OF GASES AND MATERIALS IN A 6-FT. AND 7-FT. BY 100-FT. ROTARY KILN

7 In the manufacture of Portland cement there are, roughly, three stages:

a Preparation of the raw materials, which consists of quarrying the rock and shale or clay, crushing and drying, and pulverizing the proper mixture to an average fineness of 90 to 98 per cent through a 100 mesh sieve.

b Burning or clinkering the mixture to a degree of temperature (about 2600 deg. fahr.) sufficient for the fusing of the powdered material into small greenish-black clinkers about the size of beans.

c Reducing or grinding this clinker to the required fineness, generally 95 per cent through a 100 mesh sieve.

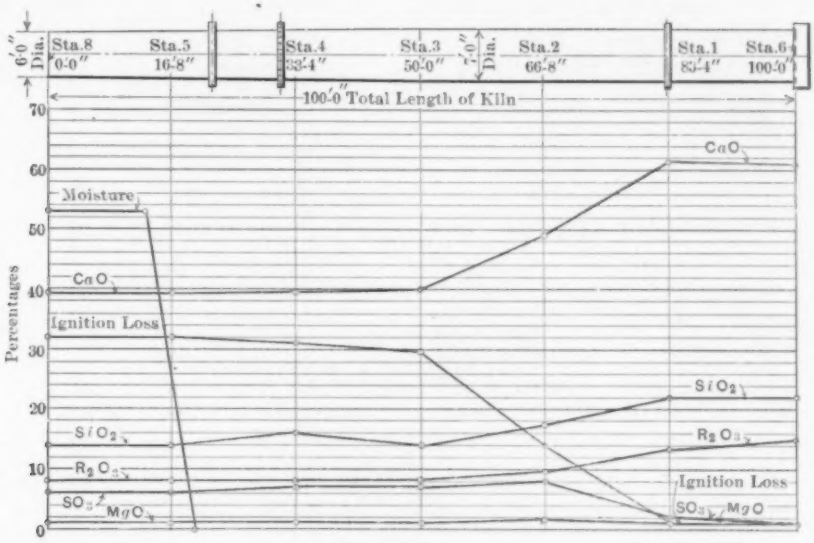


FIG. 3 CHEMICAL CHANGES IN A 6-FT. AND 7-FT. BY 100-FT. ROTARY KILN

8 Fig. 1 shows a typical installation of an 8-ft. by 125-ft. kiln. The raw mix or kiln feed enters the stack end of the kiln, either by gravity or by screw-conveyor. The kiln is lined throughout with nine inches of high refractory magnesia brick and is set at an incline of $\frac{3}{8}$ to $\frac{3}{4}$ in. per ft., which allows the material to travel slowly towards the other end of the kiln, from which it discharges into a conveyor, cooler or elevator. A mixture of air with gas, oil or pulverized coal is blown into the discharge end of the kiln by means of compressed air furnished by an air compressor or blower. When ignited, this mixture of air and combustible forms a flame or blast

of variable length which comes into contact with the feed and drives off the moisture and then the gases, after which clinkering takes place at about 2600 deg. fahr.

9 The exact temperatures at different points throughout the kiln were measured in a 7-ft. by 100-ft. kiln operating upon the wet process in which the kiln feed contained 50 per cent moisture. The

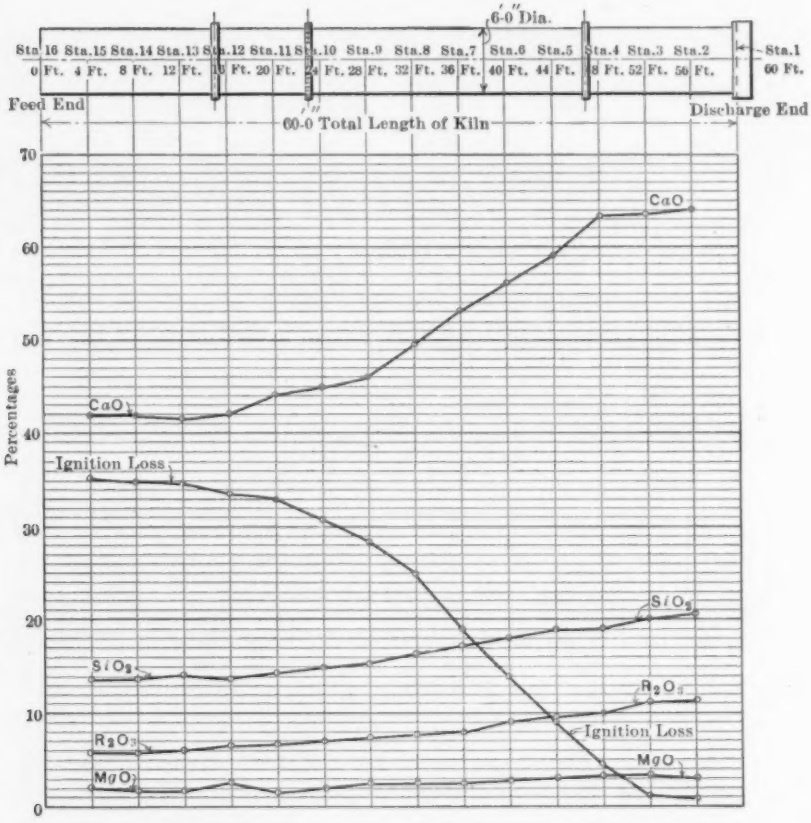


FIG. 4 CHEMICAL CHANGES IN A 6-FT. BY 60-FT. ROTARY KILN

temperatures were taken by means of a Le Chetelier pyrometer by inserting the porcelain tubes through holes previously drilled through the kiln shell and lining. The kiln revolved very slowly (about one revolution in from one to four minutes) and the temperatures of the gases were accurately determined. The temperatures of the material

FOLDER No. 1

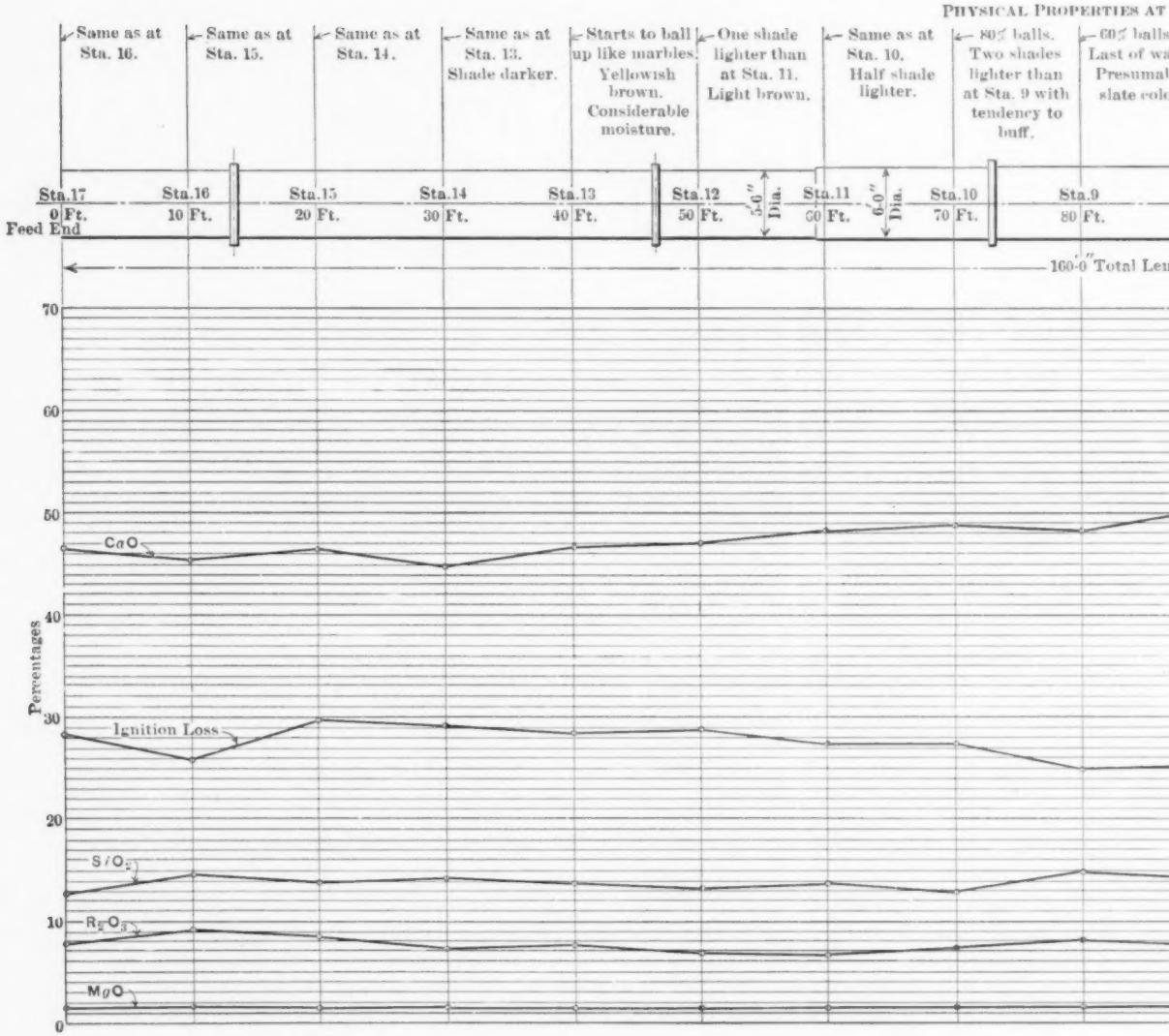
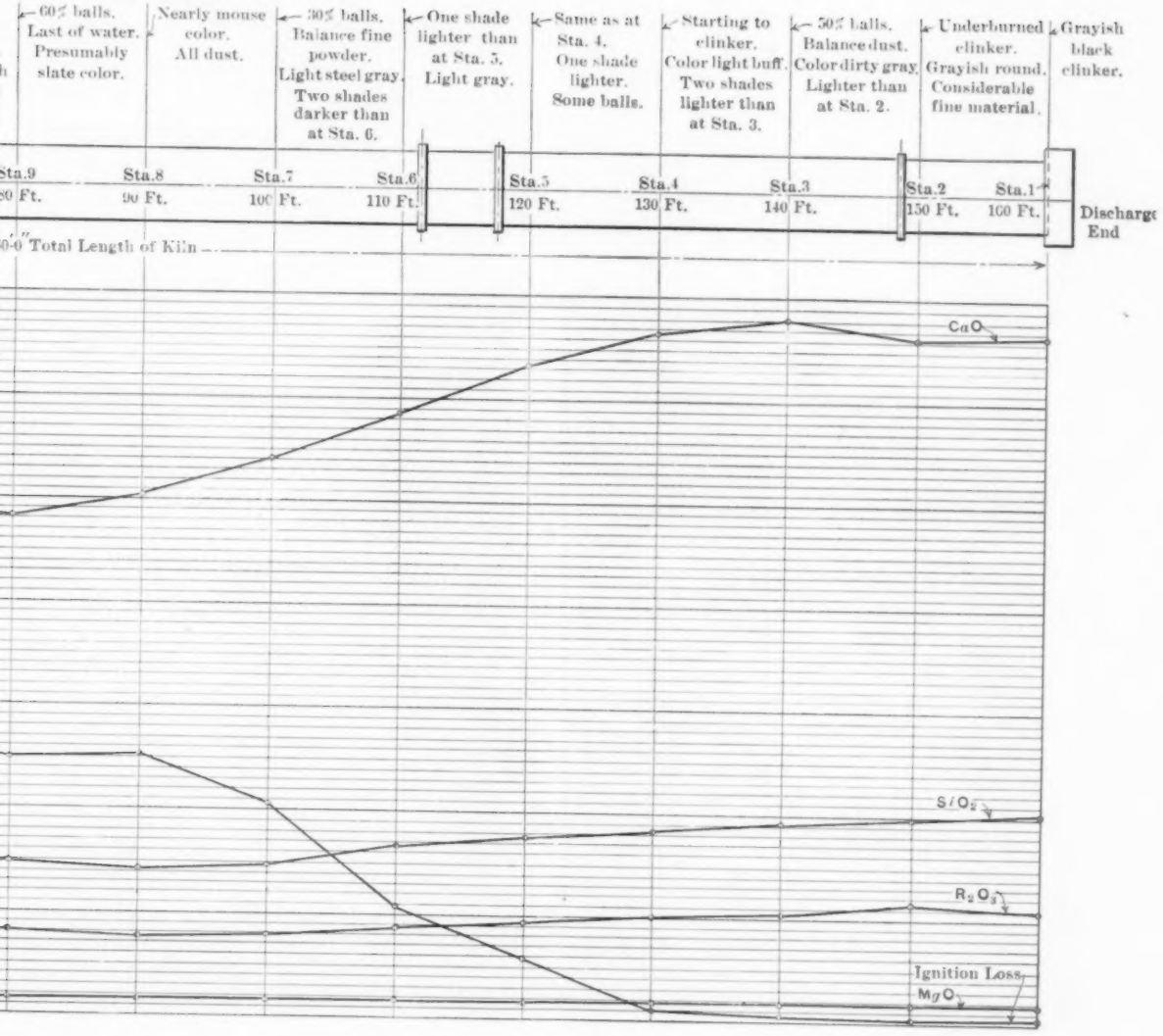


FIG. 5 CHEMICAL AND PHYSICAL CHANGES I

ROTARY KILN

OPERTIES AT DIFFERENT STATIONS


60% balls. Last of water. Presumably slate color.
Nearly mouse color. All dust.
30% balls. Balance fine powder. Light steel gray. Two shades darker than at Sta. 6.
One shade lighter than at Sta. 5. Light gray.
Same as at Sta. 4. One shade lighter. Some balls.
Starting to clinker. Color light buff. Two shades lighter than at Sta. 3.
50% balls. Balance dust. Color dirty gray. Lighter than at Sta. 2.
Underburned clinker. Grayish round. Considerable fine material.
Grayish black clinker.
Sta.9
Sta.8 90 Ft.
Sta.7 100 Ft.
Sta.6 110 Ft.
Sta.5 120 Ft.
Sta.4 130 Ft.
Sta.3 140 Ft.
Sta.2 150 Ft.
Sta.1 160 Ft.
Discharge End
Total Length of Kiln
CaO
SiO2
R2O3
Ignition Loss
MgO

HANGES IN A 6-FT. BY 160-FT. ROTARY KILN

U of M

were calculated from this data, as it was very difficult to determine the temperatures of the material without breaking the porcelain tubes. These results were plotted as shown in Fig. 2. Samples of the material were also taken for analysis at points of temperature observations, and the results plotted in Fig. 3.

10 Fig. 4 is a curve plotted by W. B. Newberry from analyses of samples taken every 4 ft. throughout the length of a 6-ft. by 60-ft. kiln. Fig. 5 is a curve plotted from analyses of samples taken from a 5½-ft. and 6-ft. by 160-ft. kiln. This curve shows that the last or upper 50 feet was comparatively useless for this diameter since there was no appreciable chemical change in that part of the kiln.

11 A list of kiln sizes, together with outputs, fuel consumptions, etc., is given in Table 1. It has been observed in practice that the

TABLE 1 KILN SIZES, OUTPUTS AND FUEL CONSUMPTIONS

Lining Inches	Diameter Feet	Length Feet	Output Bbl. per day	Fuel Consumption Lb. Coal per Bbl.
6	6	60	175	140–160
6	7	80	300	125–135
6	7	100	375	115–125
9	8	110	550	100–110
9	8	125	600	90–100
9	8	135	675	85– 90
9	8	150	700	80– 85
12	9	150	800	75– 85
12	10	175	1500	70– 80
12	12	200	2000	65– 70

diameter bears a certain relation to the length of the kiln when output and fuel consumption are considered: i.e., a 6-ft. by 60-ft. kiln produces 175 bbl. at 150 lb. coal; a 6-ft. by 100-ft. kiln produces 300 bbl. at 125 lb. coal; while an 8-ft. by 100-ft. kiln will produce 450 to 500 bbl. at 110 to 115 lb. coal. A typical and popular size just now is 8 ft. to 9 ft. by 125 ft. to 130 ft. long. This relation of diameter to length is expressed as follows:

$$L = D \times 16 \text{ (about)}$$

where L = length of kiln in ft. and D = net diameter of kiln in ft.

Actual results secured by lengthening kiln, materials and process remaining unaltered.

Size Ft.	Output Bbl.	Fuel Consumption per bbl., lb.
6 x 60	140	240
6 x 90	200	150

12 *Heat Balance.* The following example will show the method used to determine the distribution of heat in the kiln and the chemical and physical changes taking place during the burning process. The calculations are made for one barrel of cement, taking the actual analyses of raw materials and coal and determining from them the mix or kiln feed.

DATA AND DIMENSIONS

Process ..... Dry
Size of kiln ..... 8 ft. x 125 ft.
Output of kiln ..... 600 bbl. per day or 25 bbl. per h.r.
Fuel consumption ..... 90 lb. coal per bbl.

Proximate analysis of coal

	per cent
Volatile matter	38.50
Fixed carbon	52.75
Ash	7.50
Sulphur	1.50

B.t.u. per lb. = $(14{,}544 \times 0.5275 + 16{,}515 \times 0.385) + (354 \times 0.075 - 1635) =$ 12,421 B.t.u.

Area of kiln ..... 3141 sq. ft.
Area of hood ..... 185 sq. ft.
Size of stack ..... $6\frac{1}{2}$ ft. $\times$ 125 ft

TEMPERATURES

	deg. fahr.
Air entering kiln from blower	70
Air surrounding kiln (average)	70
Raw mix	60
Clinker discharging from kiln	1400
Waste gases to stack	650
Clinkering zone	2500
Temperature at which gases are liberated	1000

Specific heats

Air	0.2375
Waste gases	0.23
Limestone	0.166
Shale	0.2
Raw mix	0.2
CO_2	0.24
SO_2	0.24

Heats of combination

CaO	954.0 B.t.u. per lb.
MgO	1488.6 B.t.u. per lb.

Heats of decomposition

SO_3	1890 B.t.u. per lb.
$CaCO_3$	765 B.t.u. per lb.

ANALYSES

	per cent Rock	per cent Shale
Ignition Loss	43.34	3.00
SiO_2	1.54	66.20
Fe_2O_3	0.37	5.10
Al_2O_3	0.75	18.50
CaO	53.82	3.00
MgO	0.80	1.50

CALCULATIONS

Shale $(2.8 \times 66.2 + 18.50 \times 1.1 + 5.10 \times 0.7) - (3 + 1.5 \times 1.4) = 204.18 = n$

Rock $(53.82 + 0.8 \times 1.4) - (1.54 \times 2.8) + (0.75 \times 1.1) + (0.37 \times 0.7) = 49.55 = m$

$$\frac{n}{m} = \text{parts rock to one part of shale} = \frac{204.07}{49.55} = 4.12$$

Less 10 per cent for safety.....0.41
Parts rock to one part shale.....3.71

	Rock		Shale		
Loss	43.34 × 3.71	=	160.79 + 3.0	=	163.79
SiO_2	1.54 × 3.71	=	5.71 + 66.2	=	71.91
Fe_2O_3	0.37 × 3.71	=	1.37 + 5.1	=	6.47
Al_2O_3	0.75 × 3.71	=	2.78 + 18.5	=	21.28
CaO	53.82 × 3.71	=	199.67 + 3.0	=	202.67
MgO	0.80 × 3.71	=	2.97 + 1.5	=	4.47
					470.59

Raw Mix

		per cent
Loss	163.79 ÷ 470.59	= 34.80
SiO_2	71.91 ÷ 470.59	= 15.27
Fe_2O_3	6.47 ÷ 470.59	= 1.37
Al_2O_3	21.28 ÷ 470.59	= 4.52
CaO	202.67 ÷ 470.59	= 43.07
MgO	4.47 ÷ 470.59	= 0.95
		99.98

100 − 34.8 loss = 65.2 per cent available for cement

Finished Cement

		per cent
SiO_2	15.27 ÷ 0.652	= 23.40
Fe_2O_3	1.37 ÷ 0.652	= 2.10
Al_2O_3	4.52 ÷ 0.652	= 6.93
CaO	43.07 ÷ 0.652	= 66.00
MgO	0.95 ÷ 0.652	= 1.45
		99.88

Cementation Index

$$\frac{(2.8 \times 23.4) + (1.1 \times 6.93) + (0.7 \times 2.1)}{66 + (1.4 \times 1.45)} = 1.09$$

Hence there is no free lime in the cement made from this mixture.

HEAT BALANCE

HEAT DISTRIBUTION IN KILN PER BBL. OF CEMENT BURNED

a 77.87 per cent $CaCO_3$ to be dissociated

$0.7787 \times 600 = 466.5$ lb. $CaCO_3$

$466.5 \times 765 = 357{,}000$ B.t.u.

b 600 lb. dry raw mix to be heated from 60 deg. to 1000 deg. fahr. (temperature at which gases are liberated)

$600\ (1000 - 60) \times 0.2$ (sp. ht.) $= 112.800$ B.t.u.

c 380 lb. mix heated from 1000 deg. to 2500 deg. fahr. (clinkering temperature)

$380\ (2500 - 1000) \times 0.24$ (sp. ht.) $= 136{,}800$ B.t.u.

d 380 lb. clinker discharged at 1400 deg. fahr. loses by radiation,

$380\ (1400 - 100$ deg.$) \times 0.24$ (sp. ht.) $= 118{,}500$ B.t.u.

e Loss by radiation from shell and hood.

(1) Kiln shell

W = total loss in B.t.u. per sq. ft. per hr.

S = Coefficient of radiation through a rough steel surface = 2.77

T_1 = Average temperature kiln shell = 450 deg. fahr.

T_2 = Average temperature air = 70 deg. fahr.

B = Coefficient of construction = 6

$$W = \frac{125\ S\ (1.0077\ T_1 - 1.0077\ T_2) - 0.55\ B\ (T_1 - T_2)}{76.9}$$

$$= \frac{125 \times 2.77\ (1.0077 \times 450 - 1.0077 \times 70) - 0.55 \times 6\ (450 - 70)}{76.9}$$

$= 1738$ B.t.u. radiation per sq. ft. per hr. from kiln shell

$3141 \times 1738 = 5{,}549{,}000$ B.t.u. radiation from kiln shell per hr.

Then $\frac{5{,}430{,}000}{25} = 218{,}360$ B.t.u. radiation per bbl. from kiln shell

(2) Hood

Area = 185 sq. ft.

	deg. fahr.
Average temperature of hood	= 450
Average temperature of air	= 70
Difference in temperature	= 380

Formula $= 185 \times 380 \times 0.74 = 52{,}000$ B.t.u.

2.84 = ratio of increase of radiation for difference in temperature of 380 deg.

Then $2.84 \times 52{,}000 = 147$ B.t.u. radiation per hr. from hood

$\frac{147{,}700}{25} = 5910$ B.t.u. radiation per bbl. from hood

f Heat carried off by CO_2, etc. (temperature escaping to stock 650 deg.)

$208.8\ (650 - 70) \times 0.24$ (sp. ht.) $= 29{,}080$ B.t.u.

g Heat carried off by waste gases

Weight air required to burn 1 lb. coal = 8 lb. (approx.) Assume 1½ times theoretical air supply.

$8 \times 1.5 = 12$ lb. air required to burn 1 lb. coal
$12 \times 90 = 1080$ lb. air required per bbl.
$(650 - 70) \times 0.23$ (sp. ht.) = 133.5 B.t.u. per lb. air
Then the loss per bbl. = $1080 \times 133.5 = 144{,}190$ B.t.u.

HEAT DELIVERED TO KILN

a Heat produced by combustion of coal
B.t.u. per lb. coal assuming 1.5 times theoretical air supply = 12,421 B.t.u.
$8 \times 1.5 = 12$ lb. air per lb. coal
$12 - 8 = 4$ lb. air excess
$4\ (650 - 70) \times 0.2375$ (sp. ht.) = 550 B.t.u. absorbed per lb. coal by excess air
Then $12{,}421 - 550 = 11{,}871$ B.t.u. per lb. coal available
$90 \times 11{,}871 = 1{,}068{,}390$ B.t.u. total heat of combustion

b Heat received due to cooling of gases (CO_2 etc.) from 1000 deg. to 650 deg. fahr.
$0.348 \times 600 = 208.8$ lb. CO_2, etc.
$208.8\ (1000 - 650) \times 0.24$ (sp. ht.) = 17,520 B.t.u.

c Heat liberated by chemical reaction
$0.66 \times 380 = 251$ lb. CaO per bbl.
$251 \times 954 = 239{,}434$ B.t.u.
$0.0145 \times 380 = 5.51$ lb. MgO per bbl.
$5.57 \times 1489 = 8204.39$ B.t.u.

d Heat carried into kiln through blow pipe
1080 lb. air required per bbl.
$1080 \times (70 - 32) \times 0.2375$ (sp. ht.) = 9750 B.t.u.

SUMMARY

Heat distribution in kiln

	B.t.u. per bbl.	per cent
a Dissociation of carbonates	357,000	26.6
b Heating 600 lb. dry raw mix from 60 deg. to 1000 deg. fahr	112,800	8.26
c Heating 380 lb. mix from 1000 deg. to 2500 deg. fahr	136,800	10.00
d Loss through radiation from discharged clinker	118,500	8.8
e Radiation from shell and hood	224,270	16.8
f Carried off by gases (CO_2 etc.)	29,040	2.16
g Carried off by waste gases	144,190	10.70
Total	1,122,640	83.32
Received by kiln	1,341,322	.....
Difference unaccounted for (probably radiation)	218,682	16.45
Total	.....	99.77

Heat received by kiln

	B.t.u. per bbl.	per cent
a Combustion of coal	1,068,390	79.65
b From cooling gases	17,520	1.30
c Liberated by chemical reactions	247,662	18.46
d Delivered through air pipe	9,750	0.72
Total	1,341,322	100.13

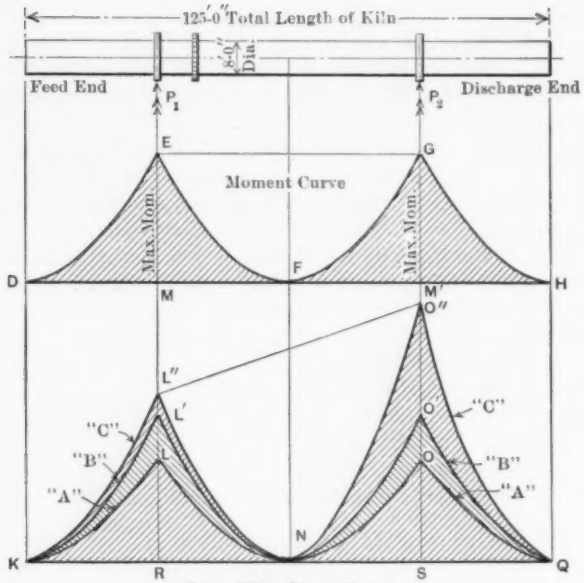


FIG. 6 STRESS DIAGRAM FOR AN 8-FT. BY 125-FT. ROTARY KILN WITH IMPROPER TIRE SPACING

From the foregoing summary, it will be noticed that only 44 per cent of the heat delivered to the kiln is theoretically required, the balance being lost through radiation, carried off by waste gases, etc.

13 *Design.* In designing a kiln it is necessary to take into consideration the weakening effect of the heat upon the strength of the steel. For this reason it is necessary so to space the riding tires or supports that the outer fiber stresses at points of maximum bend-

ing moments will be nearly equal after considering the weakening effect of the heat and the joint efficiences. Fig. 6 illustrates the effect of improper spacing of tires. This kiln, the exact duplicate of the one in Fig. 8, will carry a load only one-half as great, assuming the same factors of safety, due to the location of the points of support.

14 Were it not for the presence of heat, the design of a kiln would be comparatively simple, as riding tires could be placed as closely as desired. Fig. 7 illustrates an 8-ft. by 125-ft. kiln on three supports and the bending-moment curve when *all three* tires are touch-

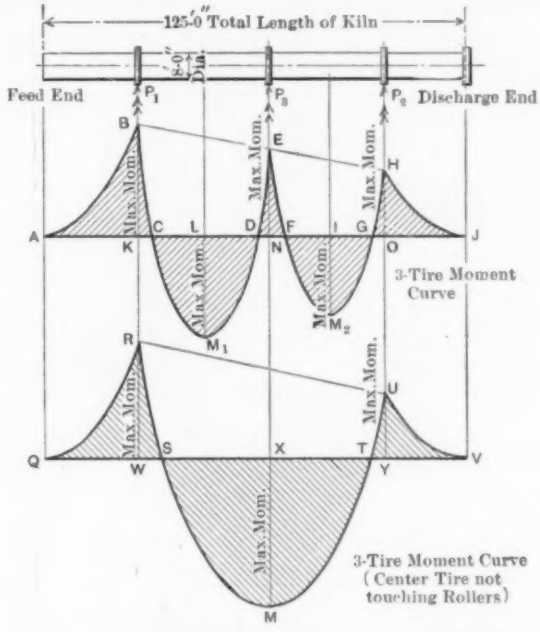


FIG. 7 STRESS DIAGRAM FOR AN 8-FT. BY 125-FT. ROTARY KILN WITH THREE SUPPORTS

ing the carrying rollers and when *only two* are touching. In operating a kiln it is very often necessary to stop its rotation for a few seconds or minutes and the intense heat gives the kiln a permanent "set." Assuming the carrying rollers to be in proper alignment, there is a portion of the revolution when the kiln is riding on *only two* tires. This obviously increases the bending moment and the outer fiber stress beyond the limits of safety, which inevitably results in the shearing of rivets or the tearing of plates. Fig. 8 illus-

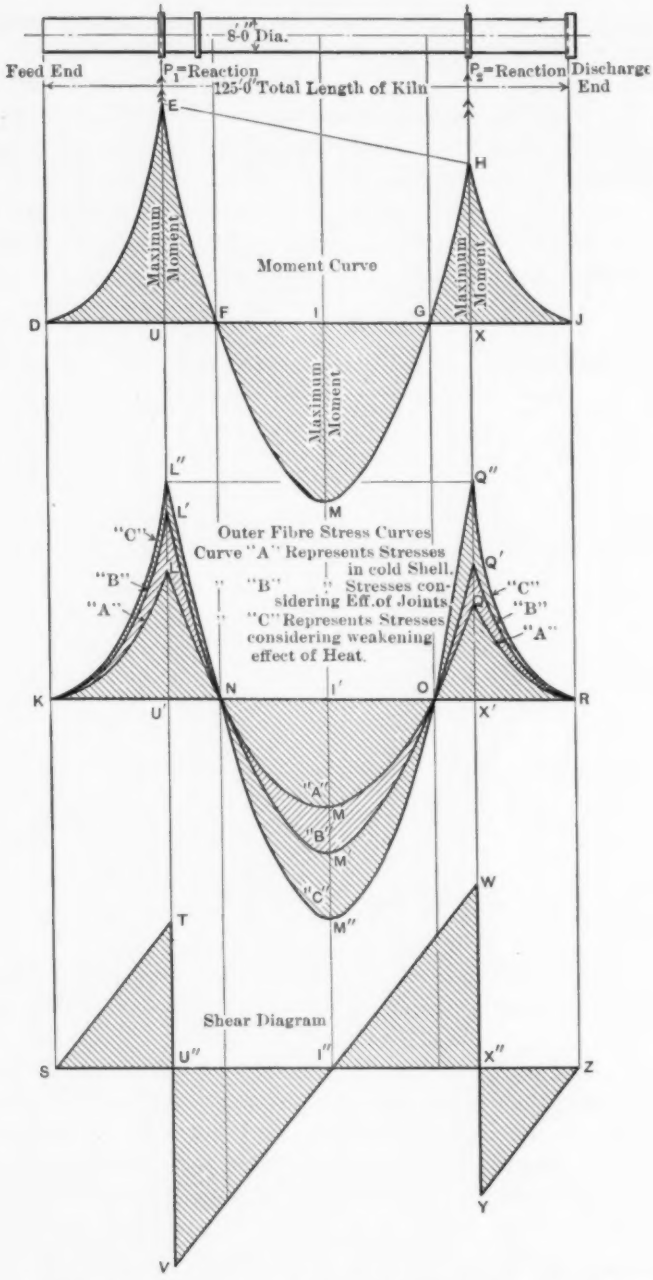


FIG. 8 STRESS DIAGRAM FOR AN 8-FT. BY 125-FT. ROTARY KILN WITH PROPER TIRE SPACING

trates an 8-ft. by 125-ft. kiln with tires properly spaced and the outer fiber-stress curves plotted. Curve *A* represents stresses in the cold shell; curve *B*, stresses after considering the efficiency of riveted joints; curve *C*, stresses considering the weakening effect of the heat. In this latter curve, the stresses *U'-L''*, *I'-M''*, and *X'-Q''* should be practically equal.

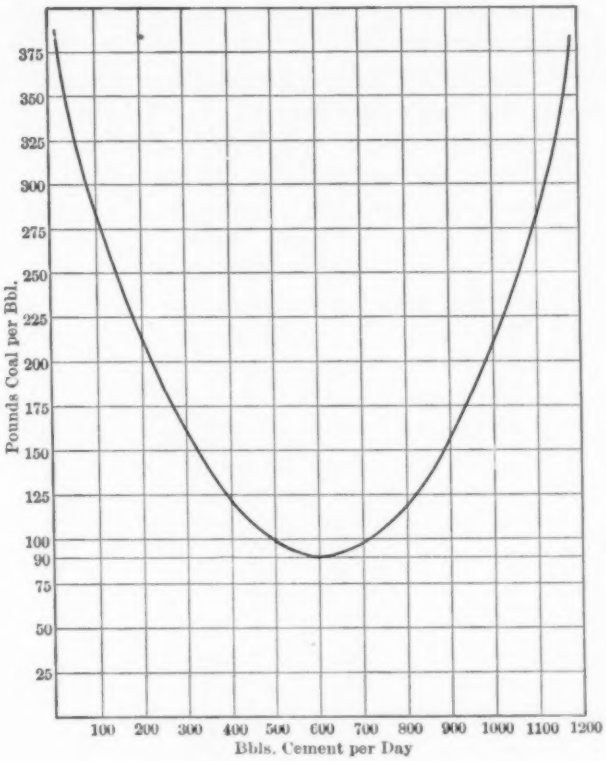


FIG. 9 FUEL CONSUMPTION CURVE

15 *The Law of Pivotal Points.* In the following paragraph an endeavor is made to apply to the rotary kiln the natural law known as the Law of Diminishing Returns or the Law of Pivotal Points. With given conditions as to materials, fuel, size of kiln and other important considerations, there is a certain production in barrels per day where the fuel consumption is a minimum. Conditions, materials and other features vary so much in d fferent locations that it is

extremely difficult to make a definite statement as to this Law of Pivotal Points for each size of kiln. In Fig. 9 is plotted the curve for an 8-ft. by 125-ft. kiln, operating upon average limestone and shale by the dry process.

16 To illustrate, if this kiln is producing 300 bbl. per day, the fuel consumption is 160 lb. per bbl.; at 400 bbl. the fuel used is 120 lb.; and the point of economical fuel consumption is 90 lb. per bbl., at which point the output is 600 bbl. per day. Beyond this production the fuel consumption increases until the kiln is literally choked.

The author hopes to be able to present more extensive information along this line in the near future.

AUTOMATIC CONTROL OF CONDENSING WATER

By B. Viola, Brooklyn, N. Y.

Member of the Society

Condensers for vacuum plants, to which this paper is chiefly devoted, employ the wet system of condensation, in which the cooling surface is that of the cooling water during its progress through the condenser.

2 In practice it is often found that the injection of cooling water is unnecessarily increased at times by incorrect operation of the condensing plant. Beside other visible indications of this mistake the temperature of the condensed water shows it most strikingly. When too much water is used the work of the air pump is made more difficult because of the air liberated from the water and if the pump is accurately proportioned for normal operation it is unable to accomplish the extra work. If the pump is large enough, the work is accomplished, but at the expense of additional steam. If a wet air pump is employed it is called upon to remove an excess of both air and water. Moreover, the excessive injection of water causes a higher pressure in the condenser. Assuming the falling water to be a liquid cylinder, the outer surface of this cylinder upon entering the hot steam takes up the heat of the steam, while the interior parts remain cooler. If, owing to the great velocity of the fall, the water column is unable to effect cooling by the interior portions of the water as well as by the exterior surface, the heat of the discharged water will be too low. But if the equalization of temperature can be brought about through the whole water cylinder during this fall the cooling power of the water can be utilized more completely.

3 The more finely the water can be divided in falling the more readily will the gases pass through it and the resistance will be diminished as the surface is increased. The division of the water

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brings about to a great extent the escape of the air contained in the cooling water. This division, however, is never fine enough; hence in counter-current condensers the real counter flow takes place only upon the surface of contact between water and steam.

4 Proper utilization of the cooling surface can only be provided for by a proper arrangement of the time of contact. For a freely

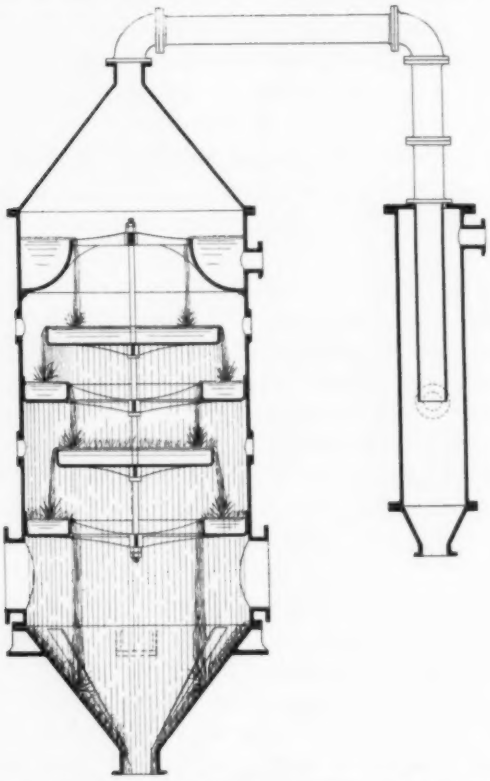


FIG. 1 CONCENTRIC COUNTER-CURRENT CONDENSER

falling body the time of falling with reference to the height of the fall is

$$t_h = \frac{v}{g} = \frac{\sqrt{2gh}}{g} = \sqrt{\frac{2}{g}} \times \sqrt{h}$$

Theoretically, therefore, the time increases as the square root of the height of the fall, but in the condenser the resistance increases with the height of the fall. Dividing the height of the fall into n parts,

we have $h = n\frac{h}{n}$. Giving to the resistance the smallest value possible and to the time of heat transfer in the condenser the highest value, the limit of the time of contact of water and heat will be

$$t_{\left(n\frac{h}{n}\right)} = n\sqrt{\frac{2}{g}} \times \sqrt{\frac{h}{n}} = \sqrt{\frac{2}{g}} \times \sqrt{nh}$$

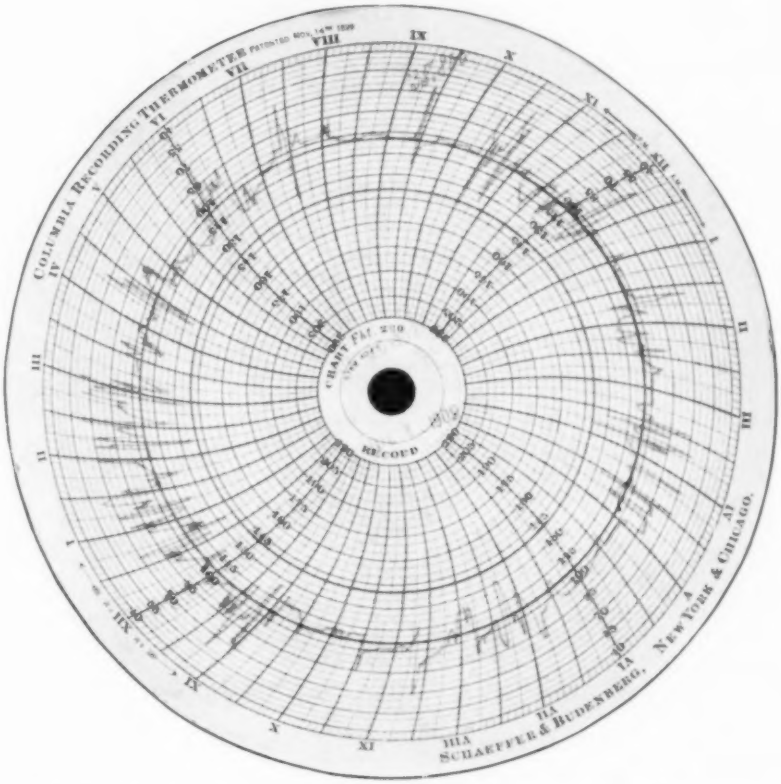


FIG. 2 AUTOGRAPHIC CHART SHOWING VARIATIONS IN TEMPERATURE UNDER HAND REGULATION

Then for every value where $n > 1$, we have $t_{\left(n\frac{h}{n}\right)} > t_h$ and in general $t_{\left(n\frac{h}{n}\right)} = \sqrt{n}t_h$. If n be taken very large and $\frac{h}{n}$ very

small, the time of contact between steam and water in the condenser will be extended proportionately since

$$\frac{h}{n}=\frac{gt^2}{2n^2}$$

5 It is necessary, however, that the partial fall $\frac{h}{n}$ be not less than the value required by the amount of steam to be condensed. From

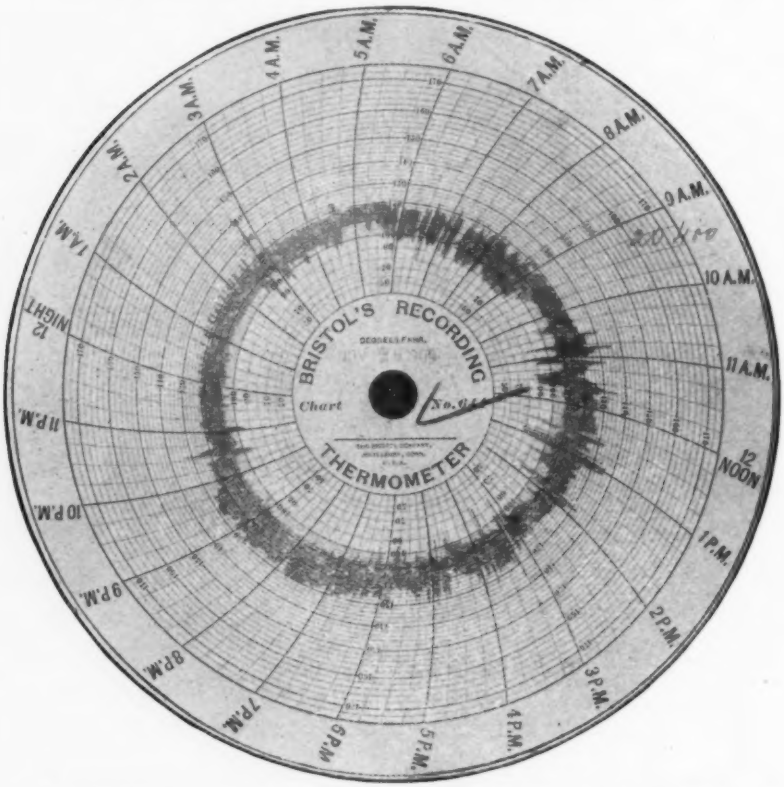


FIG. 3 AUTOGRAPHIC CHART SHOWING VARIATIONS IN TEMPERATURE WITH AUTOMATIC REGULATIONS

this calculation it is seen that the velocity of the water is governed by the height of fall and it is advisable to restrict this height in order to prevent friction resistance, if for no other reason. The smallest value of friction resistance is at $h=0$, corresponding to the condition when steam and gases flow over standing water. Plants

of such construction, however, would be too large. Moreover, water and gases being poor heat conductors when in a state of rest, it is necessary to provide for a certain degree of motion for water in a condenser.

6 Further, the quantity of steam condensed per unit of time is directly proportional to the cooling surface and a division of the cooling water is therefore equivalent to an extension of the cooling surface.

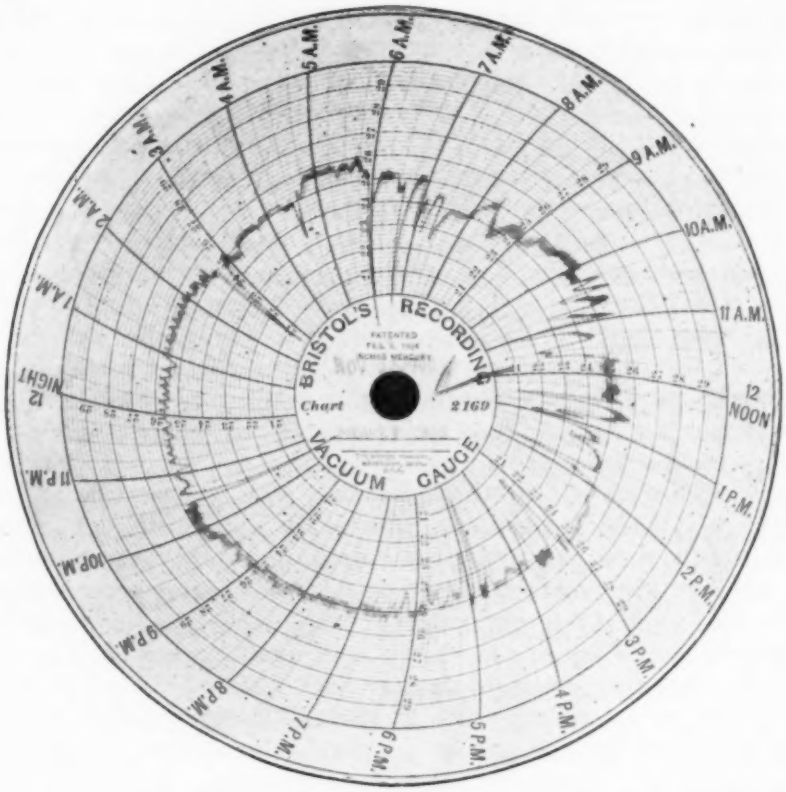


FIG. 4 AUTOGRAPHIC CHART SHOWING VARIATIONS IN VACUUM WITH AUTOMATIC REGULATION

7 Therefore, to heat condensed water quickly to the necessary temperature by direct contact with the steam, it is necessary (*a*) that the surface of the cooling water be large, (*b*) that the cooling surface change quickly, and (*c*) that the time of contact between steam and water be as long as possible.

8 To fulfill these conditions a concentric counter-current condenser such as that shown in Fig. 1 gives the best resu ts. This condenser is designed for use in connection with two or more vacuum pans. By the addition of basins or trays the duration of the contact between water and steam is raised in the ratio of $\sqrt{n}$, resulting in an economy of water. Some water will accumulate in each of these trays temporarily and the temperature is therefore equalized at each stage. The fresh supply of water will sink to the bottom of the tray, forcing the lower stratum upward on account of the difference in temperature, and the warmer water will flow over from one tray to the next. The trays in this manner form zones with the desired small difference of temperature and constitute a graduated scale from the temperature of the steam admission to that of the gas outlet. The surface of condensation depends primarily on the size of the cooling-water cylinder when the depth of the trays is greater that zero. It may even be said that these moving water columns promote the cooling and air and gases will pass at the surface of the trays even though they do not pass at the bottom. During the time the water remains in the trays it has a chance to take up more heat, which is withdrawn from the steam. It is very important that enough water be permitted to flow to secure good results, but by providing the trays with spray holes, the cooling effect of the water is considerably increased.

9 The work of the condenser varies with temperature, but in general it performs about 610 times the work of the air pump, volumetrically speaking. It is absolutely necessary to exhaust the air and gases to promote condensation, but the best air pump cannot do good work with a poor condenser, while a good condenser accomplishes at least fair results with an indifferent air pump.

10 It is very important that the right quantity of water be used to condense a certain quantity of steam. This matter is neglected in most condensing plants and some kind of automatic device should be applied to regulate the flow of cooling water. Experience has shown that the amount of water should be from 20 to 40 times that of the steam to be condensed. This is a very wide limit and in large condensing plants where immense quantities of cooling water are required some controlling device should be applied from an economical standpoint, so that water will not be wasted.

11 The first step is to decide at what temperature the water should be discharged from the condenser. This, of course, bears a certain relation to the steam to be condensed. The description

which follows is a record of experiments made and results achieved with a condensing plant used in connection with vacuum pans. I first applied a recording theremometer in the tail pipe of a concentric barometric condenser for the purpose of controlling the cooling water by hand operation of the inlet valve; but it was very difficult to convince the operator of the condenser that it was not necessary to let the cooling water run off cold, because it is generally believed the colder the water the higher the vacuum. The variations in temperature under hand control can be seen in Fig. 2. After experimenting for some time it was found very difficult to secure good results by hand regulation because of the continuous changes which must be made to meet conditions as the number of pans connected to the condenser is increased or decreased. This led to the application of an automatic temperature controller in the tail pipe connected to the compressed-air inlet valve, by which it was found possible to feed the condenser approximately in proportion to the steam or vapor which had to be condensed, resulting in a better average temperature of the condensed water and only slight variations.

12 The temperature of the condensed water at a given time may be compared with the corresponding vacuum secured by automatic regulation from Figs. 3 and 4. By maintaining a nearly equal temperature of the condensed water, a very uniform vacuum was obtained except at periods when a new vacuum pan was charged. The most striking changes will be noticed at 10.30 p.m., 2.45 a.m. and 6 a.m. in Fig. 4. At the same periods the controlling thermometer shows a considerable fall in temperature, due solely to the temperature of the air drawn in from a newly charged pan which was filled with air at the temperature of the operating room. The vacuum pump draws it into the condenser, as a consequence of which the vacuum drops until all the air is drawn out and the new pan reaches an equilibrium with the vacuum the pump is able to keep up in connection with the other pans. During this period of low vacuum the water is shut off by the controlling thermometer and no cooling water whatever enters the condenser. Since the other pans are boiling, the temperature in the condenser increases until the water left in the trays is raised to the temperature limit, when the controlling thermometer opens the inlet valve and the cooling water is again admitted.

13 The fact that the higher vacuum is not obtained with an excess of cooling water can be seen from Figs. 3 and 4. Between 4.45 and 5.15 a.m., when according to the chart the lowest temperature was

110 deg. and the highest 120 deg., the corresponding vacuum was 25.8 in. The charts shown in Figs. 5 and 6 show even higher vacuums with a temperature of 120 deg., the same vacuum pump and the same number of vacuum pans being used in each case.

14 The construction and operation of the automatic controlling device by which the foregoing results were obtained is shown in Fig.

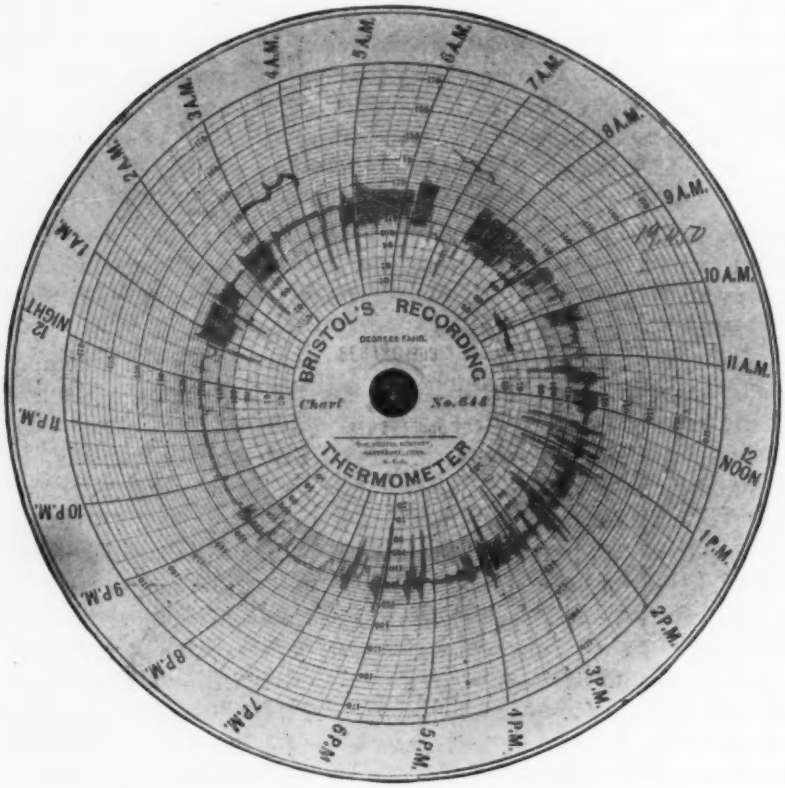


FIG. 5 AUTOGRAPHIC CHART SHOWING VARIATIONS IN TEMPERATURE WITH AUTOMATIC REGULATION

7. The stem *B* expands and contracts in response to temperature changes of the condensed water. The pipe *E* supplies compressed air of about 15 lb. pressure to the valve mechanism of the controller, from which it passes through pipe *G* to the water-inlet valve, where it actuates a regular globe valve *D* by means of a diaphragm motor which

replaces the ordinary hand wheel. The temperature of the condensed water affecting the stem of the controller permits all, a part, or none of the compressed air to pass to the diaphragm-motor valve and consequently the latter is very nearly closed, partly open, or wide open according to the conditions prevailing, allowing the proper flow of water to the condenser.

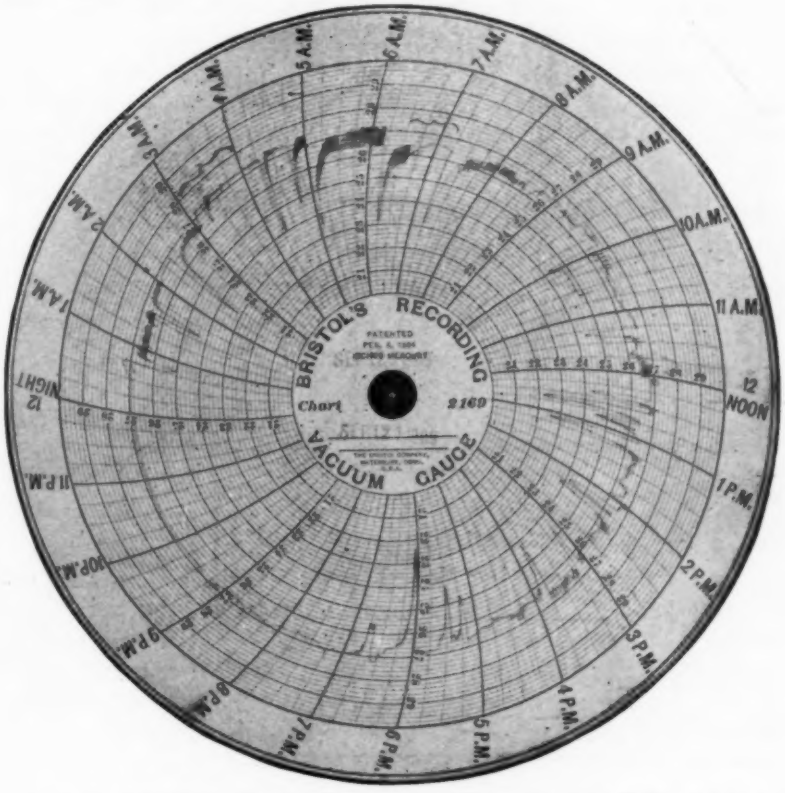


FIG. 6 AUTOGRAPHIC CHART SHOWING VARIATIONS IN VACUUM WITH AUTOMATIC REGULATION

15 Let us assume that the vapor pipe has been delivering an amount of vapor to the condenser which requires an amount of water that will flow through the inlet valve when the latter is exactly half open. Should more vapor enter the condenser, however, the temperature of the water passing through the tail pipe would, of course, start to rise, but as soon as the controller stem was affected by this higher temperature it would expand and cause the valve mechanism of the

controller to allow less air to flow to the water-inlet valve. Consequently this water valve would act to allow more water to pass through and thus maintain the proper proportion between the volumes of

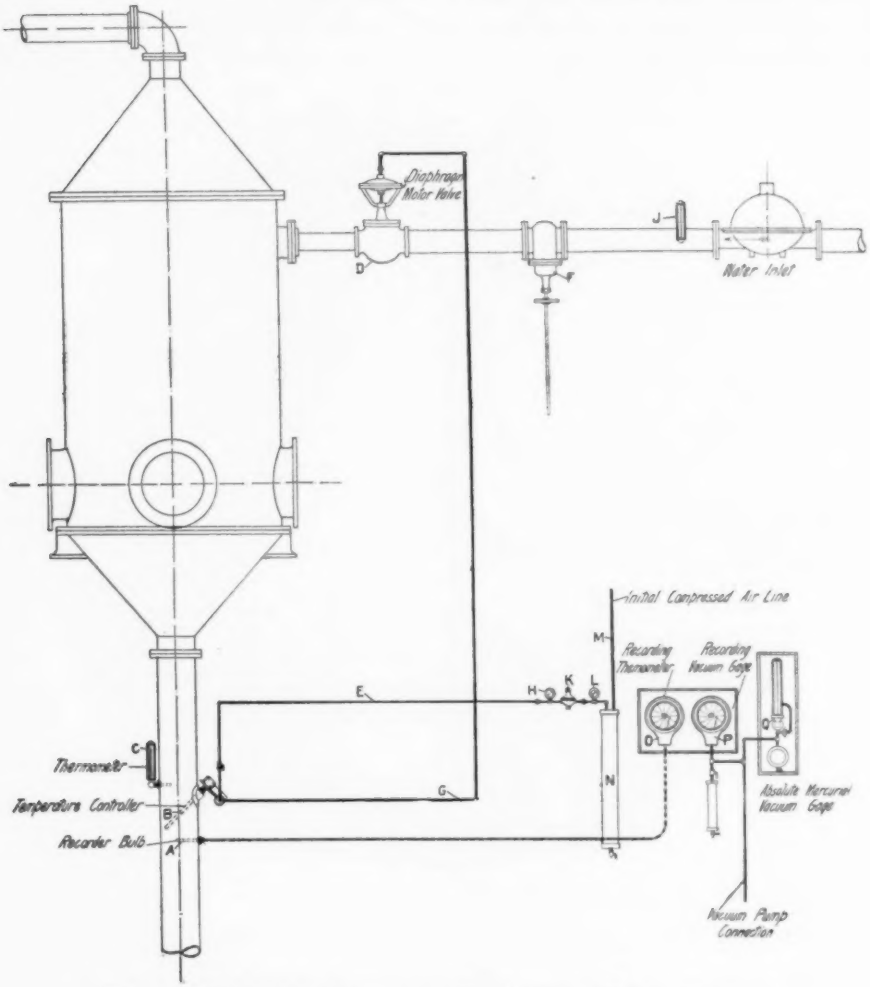


FIG. 7 DIAGRAM OF AUTOMATIC CONDENSING WATER CONTROL

vapor and cooling water. The auxiliaries are, J, a thermometer to indicate the temperature of the cooling water; F, a hand valve for emergency use; C, a thermometer to indicate the temperature of the water passing through the tail-pipe and serving as a check on the recording

thermometer, which has its bulb at A and its dial at O; and P, a recording vacuum gage, checked by a mercurial absolute-pressure gage, Q. The initial supply of compressed air, M, which ultimately operates the diaphragm-motor valve on the water inlet, enters the trap N, where any moisture or sediment is separated and which also acts as a storage reservoir. Gage L shows the initial pressure of the compressed air supply, which is reduced by passing through reducing valve K to the pressure required for the controller, as shown by gage H.

16 Fig. 8 shows the very simple construction of the controller by means of a section through A-B. The stem S is in two parts, the outer brass tube and the inner rod. The latter is composed of a material

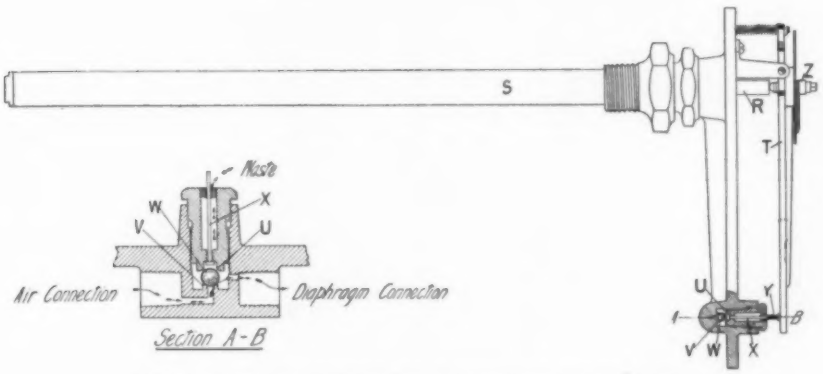


FIG. 8 CONSTRUCTION OF AUTOMATIC CONTROLLER

which has an extremely low ratio of expansion and therefore remains of almost constant length, regardless of temperature, while the outer brass tube elongates or contracts in response to temperature changes. The tube and rod are so arranged that they are always in intimate though frictionless contact at the extreme end. R shows the extension of the inner non-expanding member, which recedes when the outer tube elongates and advances when the outer tube contracts.

17 The movement thus obtained at R is transmitted by the lever T to the valve mechanism shown in cross section. This is simply a three-port valve, shown more clearly in the section A-B, and all ports are controlled by the one ball W. The first port, V, is the inlet for the compressed air supply; another port, the second, connects with the line leading to the diaphragm-motor valve; the third port is the outlet for the air which has served its purpose by actuating the diaphragm-motor. The position of the ball W depends upon the stem

X, which is in contact with the lever T at Y. When extension R moves outward due to the drop in temperature at S, the lever T is raised, and the stem X allows the ball W to leave port V where the compressed air enters. The air can then flow through the second port to the diaphragm-motor valve, which starts to close. If a higher temperature acts upon S, expanding the outer tube and turning extension R inward, lever T forces stem X against ball W. The latter then

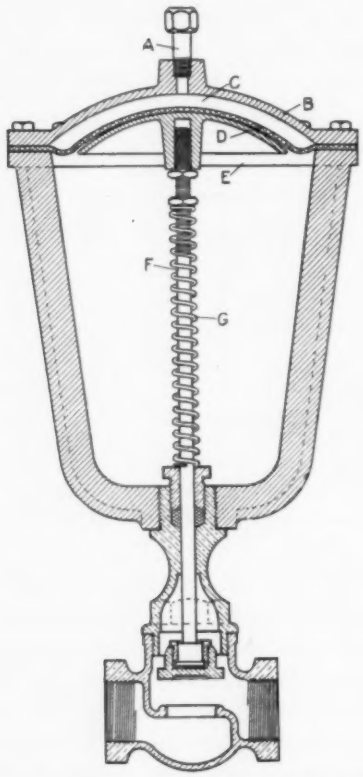


FIG. 9 SECTION THROUGH DIAPHRAGM-MOTOR VALVE

closes port V, shuts off the air supply, and also opens the third port so that the air can escape from the diaphragm motor. The diaphragm-motor valve consequently begins to open.

18 In actual practice, the diaphragm-motor valve is only completely closed or wide open if the conditions are such that the tendency is toward a sudden change in temperature. The controller is so sensitive that the diaphragm-motor valve is constantly throttling,

that is, letting in just a little more or a little less water as may be required properly to effect the condensation.

19 If it is desired to change the setting of the controller, to maintain a higher or a lower temperature, it is necessary simply to apply a key to the port Z and by turning it one way or the other require extension R to travel more or less before the controller operates.

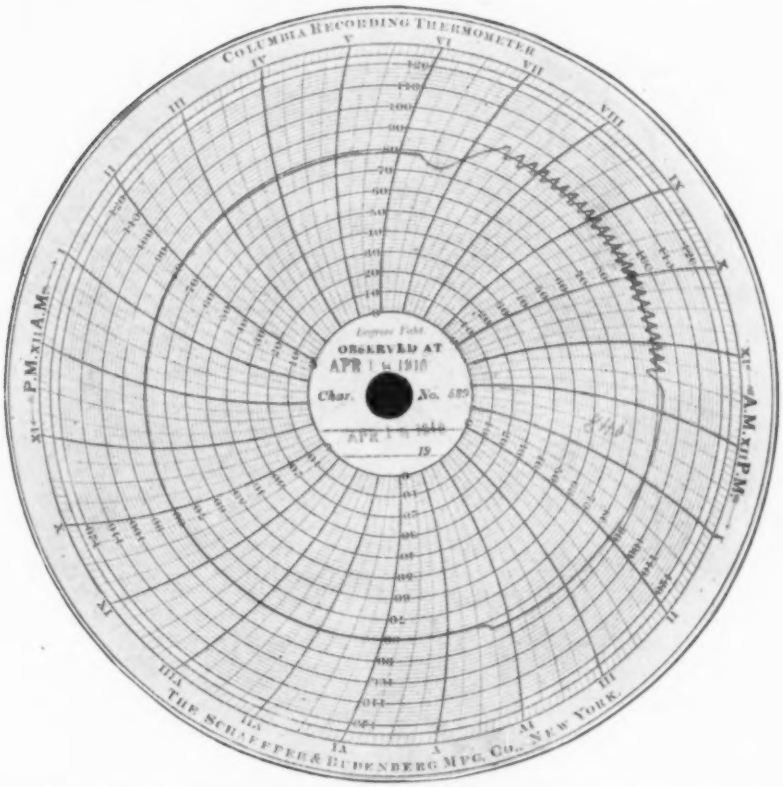


FIG. 10 AUTOGRAPHIC CHART SHOWING AUTOMATIC TEMPERATURE REGULATION OF A SURFACE CONDENSER

20 Fig. 9 shows the diaphragm-motor valve in section. This valve has a regular globe body, with a seat and disc arranged in the usual manner, but with a sliding stem that is operated by the diaphragm motor. The compressed air enters chamber C at opening A. This chamber is composed of a cast-iron top B and a rubber bottom D, called the diaphragm. The air pressure forces the diaphragm down-

ward and the saucer *E* thus forces the stem *F* downward while compressing the spring *G*. When the air pressure is relieved the spring opens the valve again.

21 This installation has been the means of saving about $400 monthly for over a year in the charges for city water which was formerly wasted.

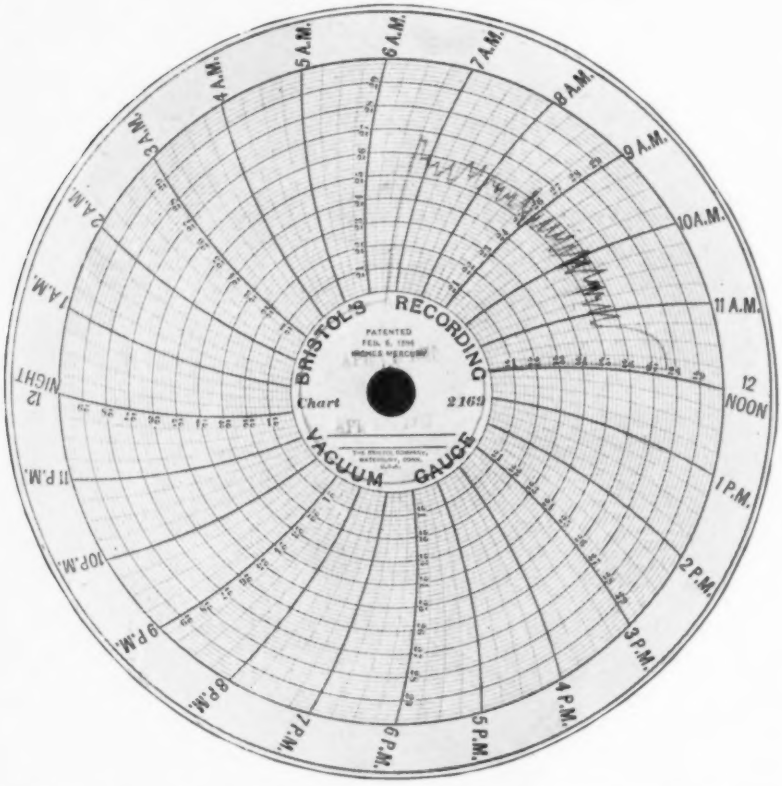


FIG. 11 AUTOGRAPHIC CHART SHOWING AUTOMATIC VACUUM REGULATION OF A SURFACE CONDENSER

22 In Figs. 10 and 11 are shown autographic charts taken from a temperature controller and applied to a single surface condenser connected to a vacuum drier. With this installation very good results have been obtained and the quantity of cooling water has been decreased from 35 to 50 per cent over that required with hand regulation.

23 The question arises as to whether, with an equipment as described above, it is possible to attain a cooling-water rate near enough to that called for by theoretical formulae to be economical. Records of the water consumption as determined by meter readings show results very close to theoretical and indicate great savings over the water required with former conditions of operation.

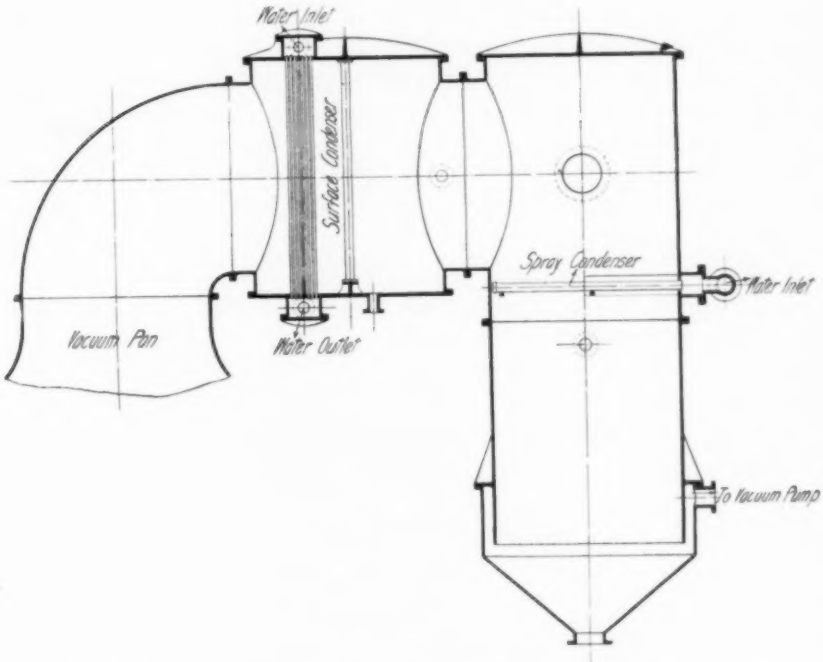


FIG. 12 COMBINED SURFACE AND SPRAY CONDENSER AS APPLIED TO A VACUUM PAN

24 In sugar refineries, combined surface and spray condensers are usually employed which are similar to that shown in Fig. 12. The vapor from the vacuum pan passes the surface condenser first for the reason that in case the vapor carries along some sugar particles, they will be condensed at that point and can be used to dissolve the raw sugar. In this way nothing is lost. The remainder of the vapor passes to the spray condenser, is condensed and flows down the tailpipe.

25 Condensers of this type require a considerable quantity of water and permit of great waste when no controlling device is employed.

Even where salt water is used this waste should not be permitted because it increases materially the quantity of steam necessary for the operation of the pumps. In many cases where salt water is now used on account of the great quantity required, the use of a well-constructed counter-current condenser operated economically with a precise temperature controller may make possible the use of sweet water. This is a decided advantage where the condensed water can be used for boiler-feed purposes. I would not recommend its use in the same condition as when it flows from the tail-pipe, in spite of the assertion that condensed water does not contain any impurities carried over from the source of the vapor. In steam engine plants for instance, it is well known that condensed steam contains oil, etc., and it is never pumped direct to the boilers. It has been determined from frequent analysis that condensed water from vacuum plants does not contain sugar or acid from the evaporation of acid liquor, but the so-called untraceable losses in vacuum plants must necessarily be looked for in the condensed water. The quantity of water used is so great that the presence of these impurities is not noticeable, but practical experience has proved them to be present. However, when this condensed water at a high temperature (110 to 120 deg.) is passed through an efficient water-purifying plant, there is no doubt that it may safely be used for boiler feed.

THE UNITED STATES BATTLESHIP NORTH DAKOTA

At a joint meeting of a number of engineering societies in and about Boston, held at the Hotel Somerset in that city on Friday evening, January 21, 1910, C. B. Edwards, chief engineer of the Fore River Shipbuilding Company, gave an illustrated lecture descriptive of the battleship North Dakota, which was at that time the latest addition to the United States Navy, and which was designed and constructed under the direction of Mr. Edwards.

FIG. 1 LAUNCHING OF THE U. S. BATTLESHIP NORTH DAKOTA, NOVEMBER 10, 1908

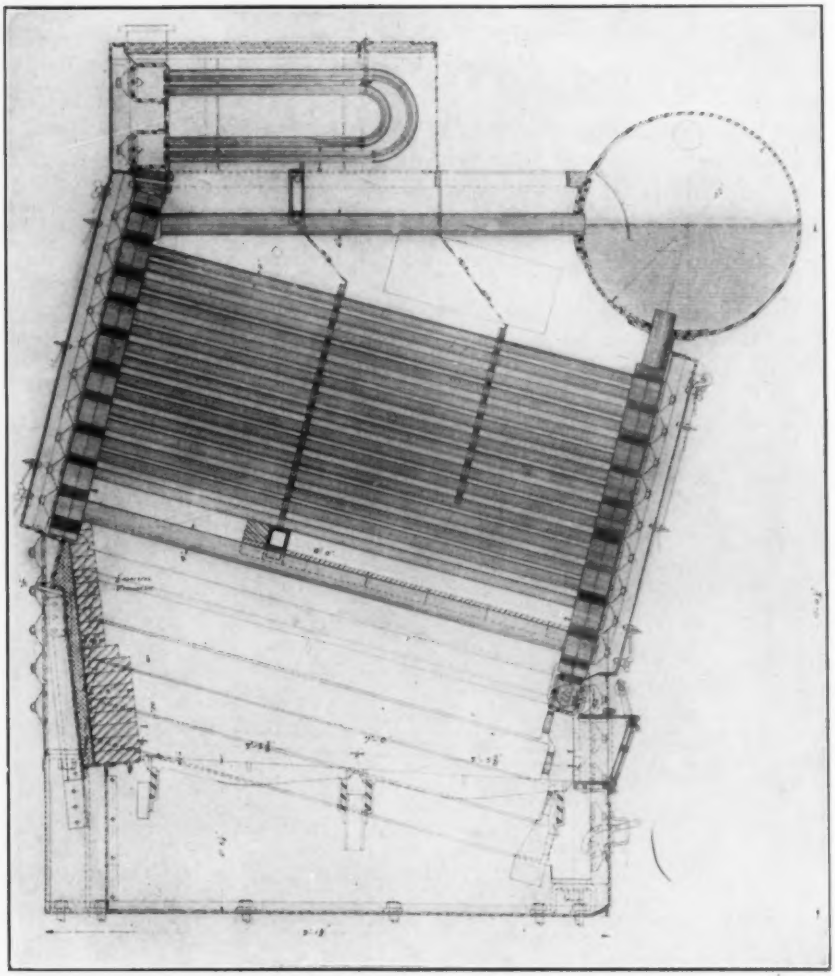


FIG. 2 SECTIONAL VIEW OF BABCOCK AND WILCOX BOILERS

These boilers are fitted with superheaters for the use of oil fuel in conjunction with coal. Grate surface, 1439 sq. ft.; heating surface, 55,800 sq. ft.; superheat surface, 6194 sq. ft.

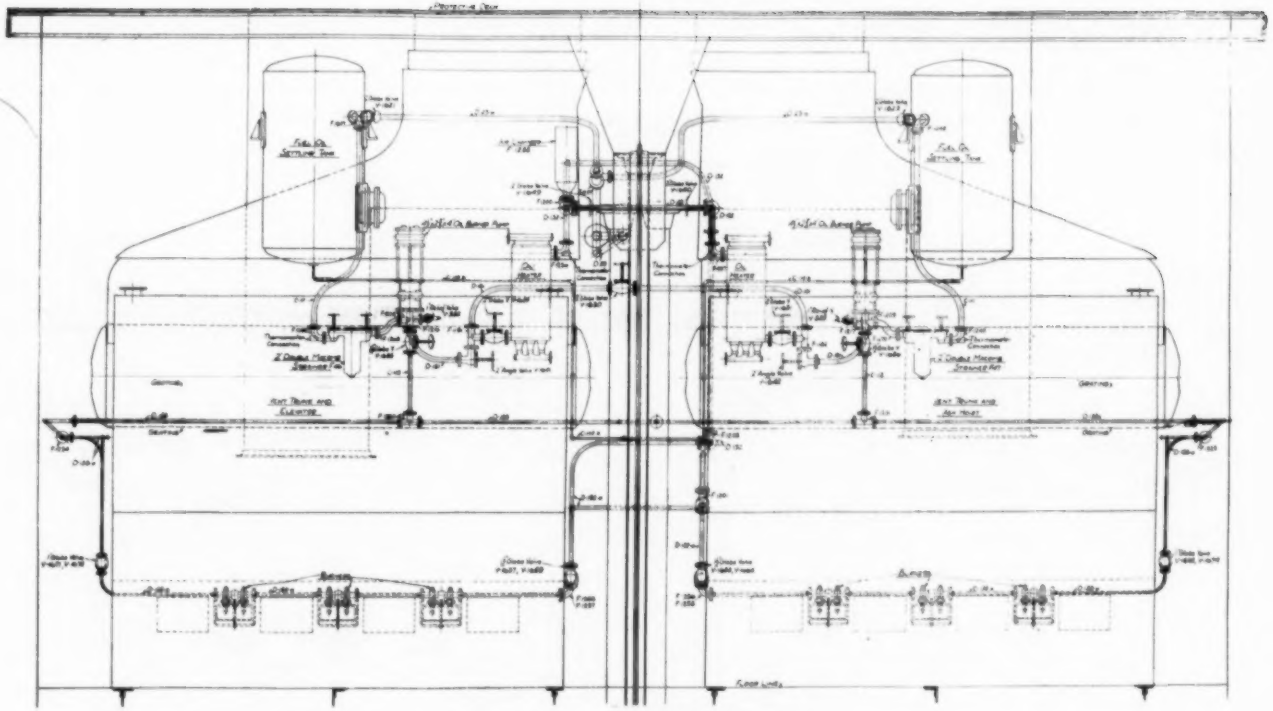


FIG. 3 SECTION THROUGH FIRE ROOM

This view shows the fuel oil arrangement of Schutte-Koerting type. The oil is injected into the furnace under a pressure of 150 lb. per sq. in. and at a temperature of 250 deg. over the top of the coal fire. This facilitates the raising of steam and the fuel oil carried in the inner bottom increases the cruising radius of the vessel.

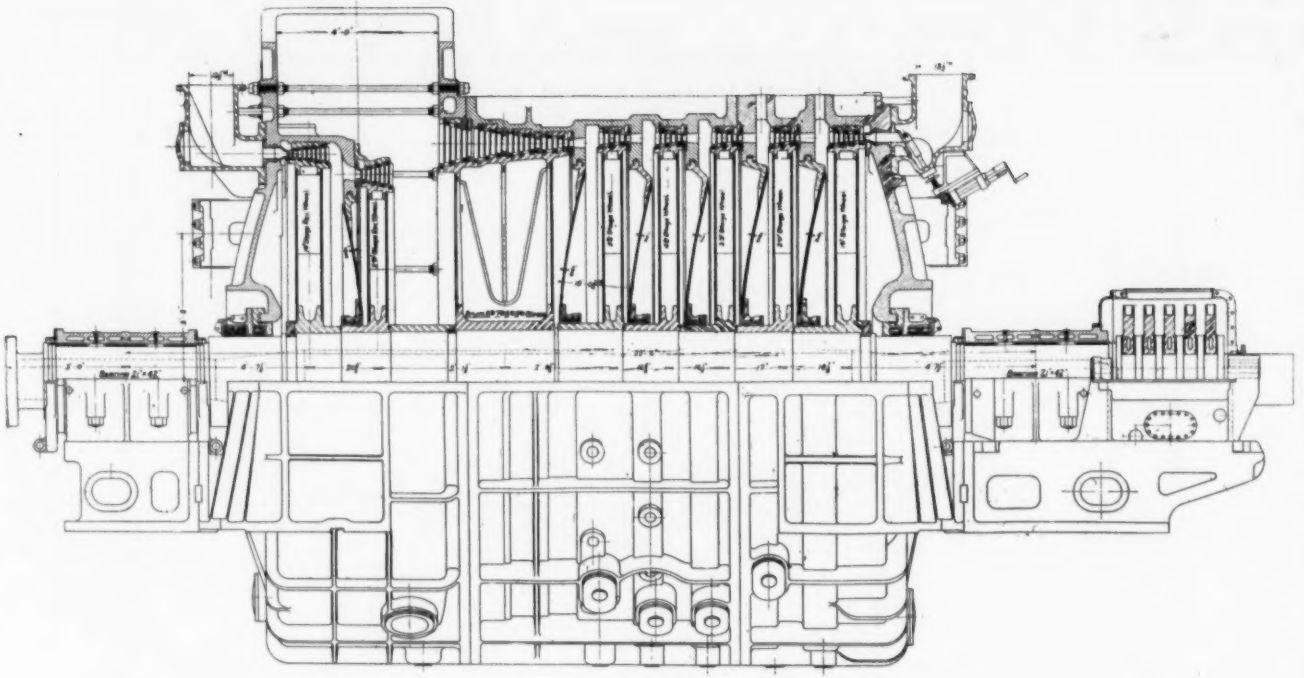


FIG. 4 SECTIONAL VIEW OF ONE OF THE MAIN TURBINES

It will be observed that this turbine is arranged for both ahead and astern power in one shell, or casing, with a common exhaust connection. For ahead motion, steam is admitted at the forward end just above the thrust bearing and passes through nine pressure stages. The rotating buckets of the first five stages are attached to the periphery of their respective wheels, while the last four are mounted upon a drum which is so proportioned that the steam pressure balances a portion of the thrust exerted by the propeller, the thrust block taking the remainder. For astern motion, steam is admitted to the after end of the turbine and passes through two turbine stages to the exhaust.

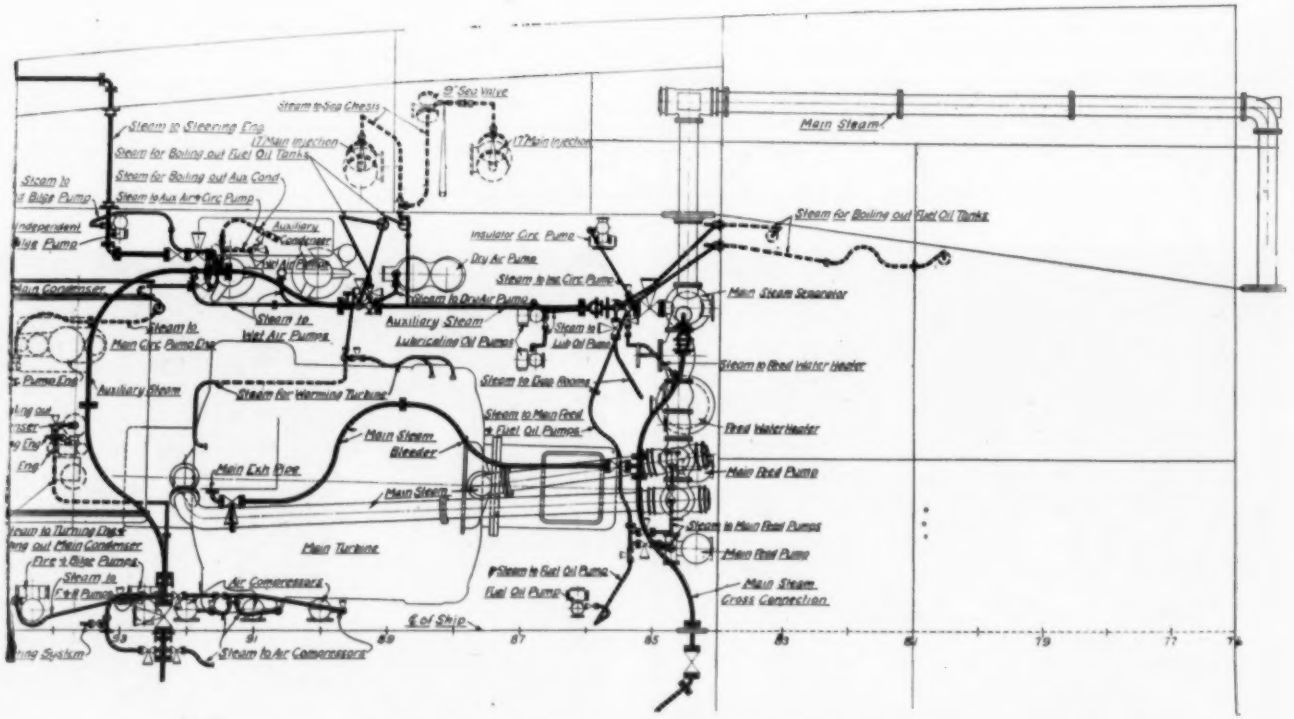


FIG. 5 PLAN VIEW OF ONE OF THE ENGINE ROOMS

The engine rooms are located directly opposite and just forward of the magazine, making it necessary to carry the main steam pipes from the boiler room around the magazine. Each engine room measures approximately 44 ft. by 25 ft. by 25½ ft., and in this contracted space about 35,000 h.p. has been obtained from the main machinery alone.

FIG. 6 FORWARD END OF ONE OF THE TURBNES

This view is taken looking aft from the working platform in the forward part of the engine room and shows the forward end of one of the main turbines, together with the main steam and exhaust pipes. In the background may be seen some of the auxiliary piping, which presents a somewhat congested appearance.

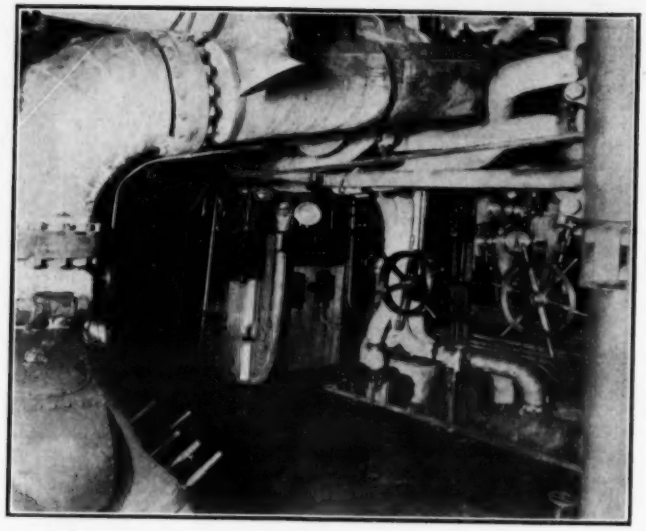


FIG. 7 WORKING PLATFORM FORWARD OF ONE OF THE TURBINES

This view is taken in the opposite direction to that of Fig.4 and shows the hand wheels for operating ahead and astern throttle valves, together with the mechanical telegraphs, telephone connections, gages, ventilating system, etc.

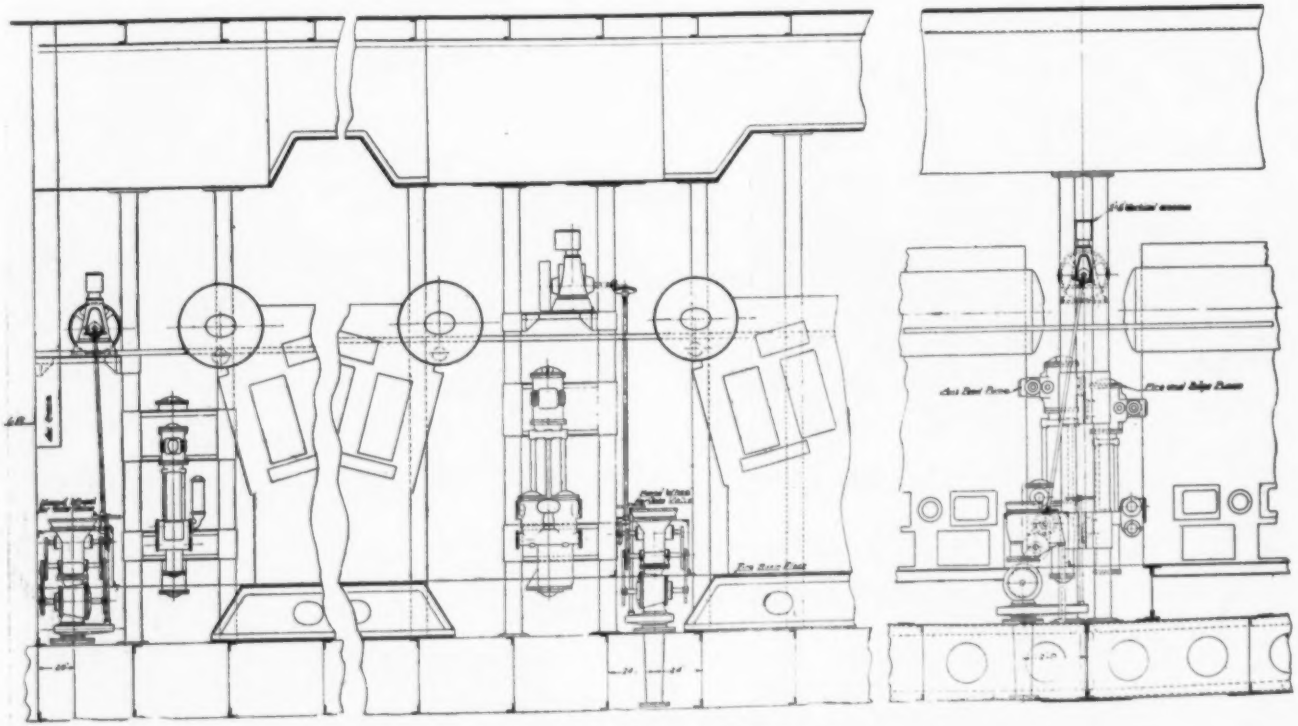


FIG. 8 ASH EXPELLER MECHANISM

This is a new departure in naval engineering by which the ashes are discharged through the bottom of the ship. They are put through a crusher and expelled by means of compressed air at 95 lb. pressure.

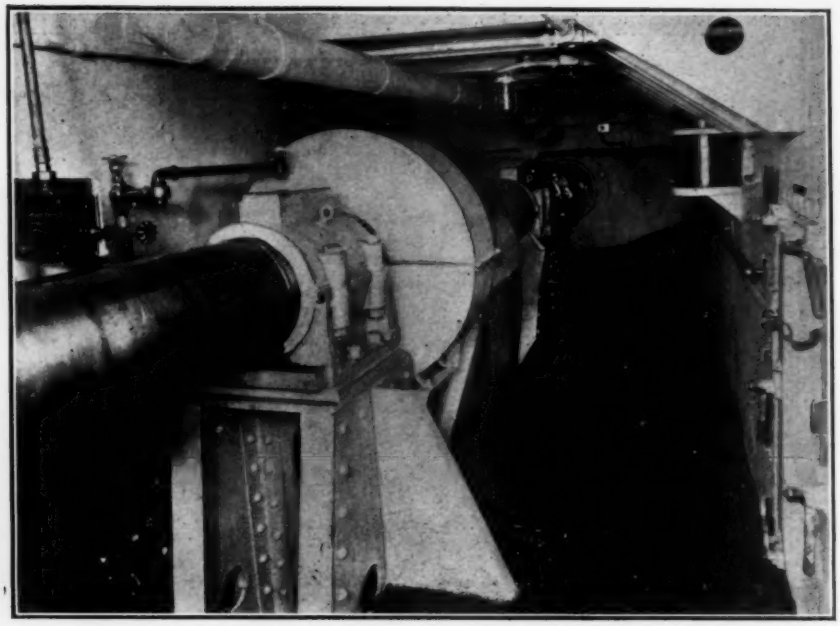


FIG. 9 TORSION METER FOR MEASURING TWIST OF SHAFT AND HORSEPOWER TRANSMITTED

The shaft is 14 in. in diameter with an 8-in. bore. The twist is about one degree in a length of 8 ft. when transmitting full power.

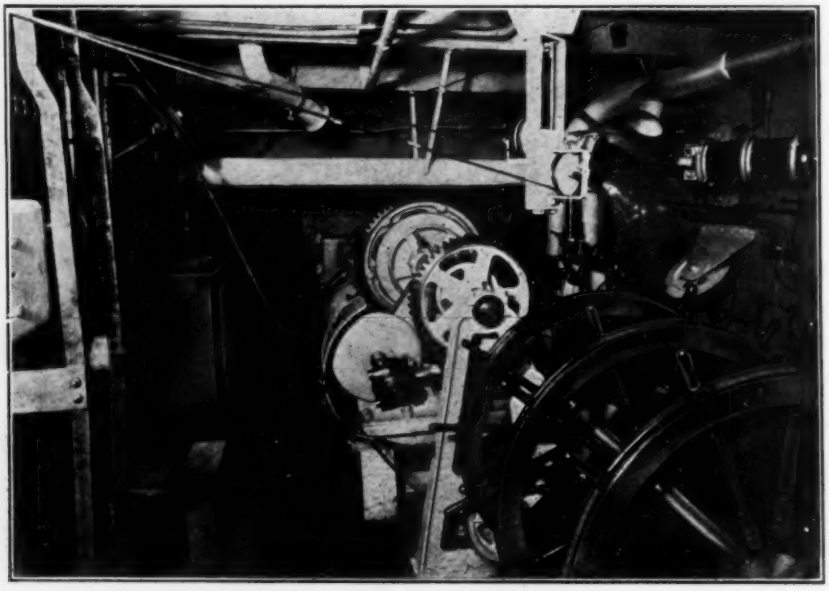


FIG. 10 STEERING-ENGINE ROOM

Wire ropes transmit the motion from the wheels on the bridge and in the conning towers to the drums for operating the steam steering gear. The large wheels provide means for steering by hand.

A number of the lantern slides used to illustrate Mr. Edwards' lecture are reproduced herewith, together with descriptive captions taken from his remarks as the pictures were thrown on the screen.

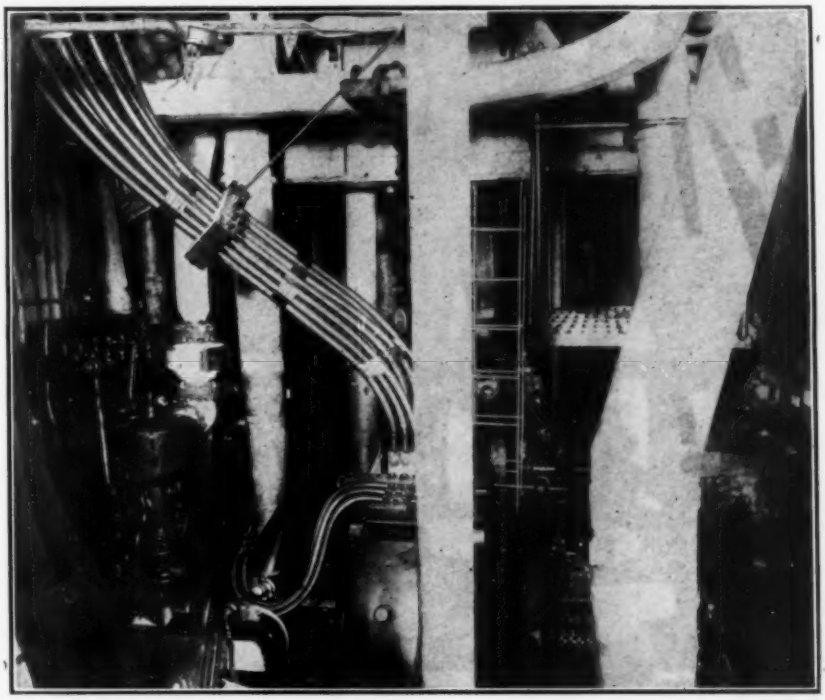


FIG. 11 THE DYNAMO ROOM

Four 300-kw. turbo-generating sets are crowded into a space of about 20 ft. by 42 ft.

The North Dakota is 518 ft. in length and 85 ft. in width, and has a draft of 26 ft. $10\frac{3}{4}$ in. She has a normal displacement of 20,000 tons and contains approximately 1950 tons of machinery. There are 431 watertight compartments, not including the inner bottom, besides many other compartments which are non-watertight. Practically all of the auxiliary machinery was tested under approximately normal operating conditions, and the water rates established before installation.

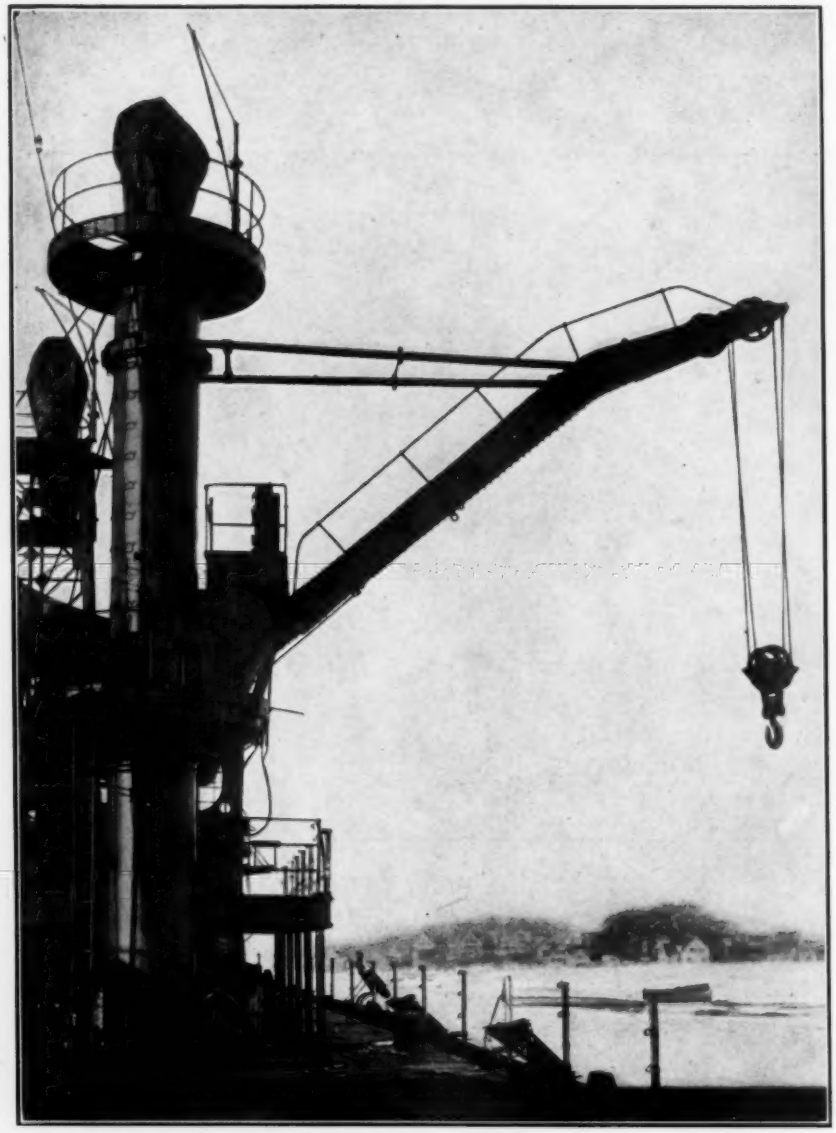


FIG. 12 CRANE FOR HOISTING BOATS ABOARD

This crane is operated by electric motors and has a capacity of 33,000 lb. Note the search'ight on top of the crane post and the one on the bridge. There are four 36-in. and four 60-in. searchlights on the ship.

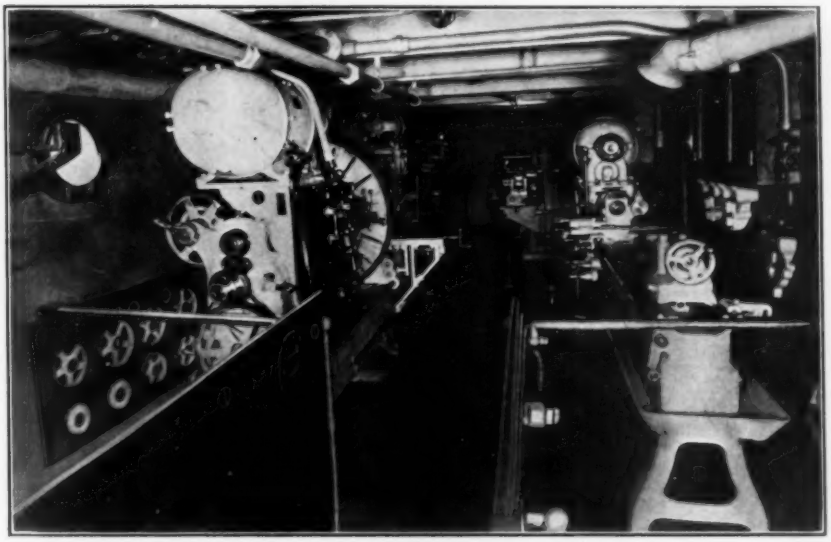


FIG. 13 WORKSHOP

Each tool is provided with individual motor drive and extensive repairs can be made on board the vessel.

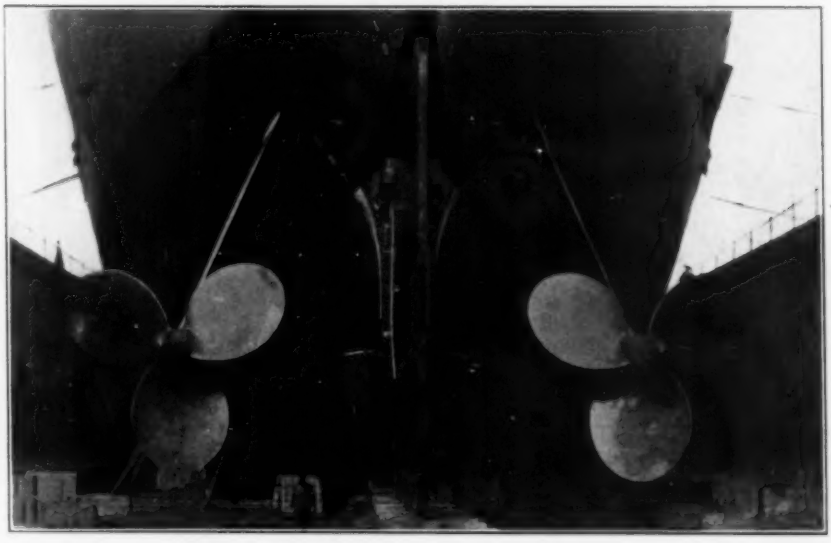


FIG. 14 STERN VIEW SHOWING PROPELLERS

The propellers are made of a new alloy known as Monel metal, containing about 60 per cent of nickel. The working surfaces are accurately planed to a true screw for a pitch of 10 ft. 4 in. and a diameter of 13 ft.

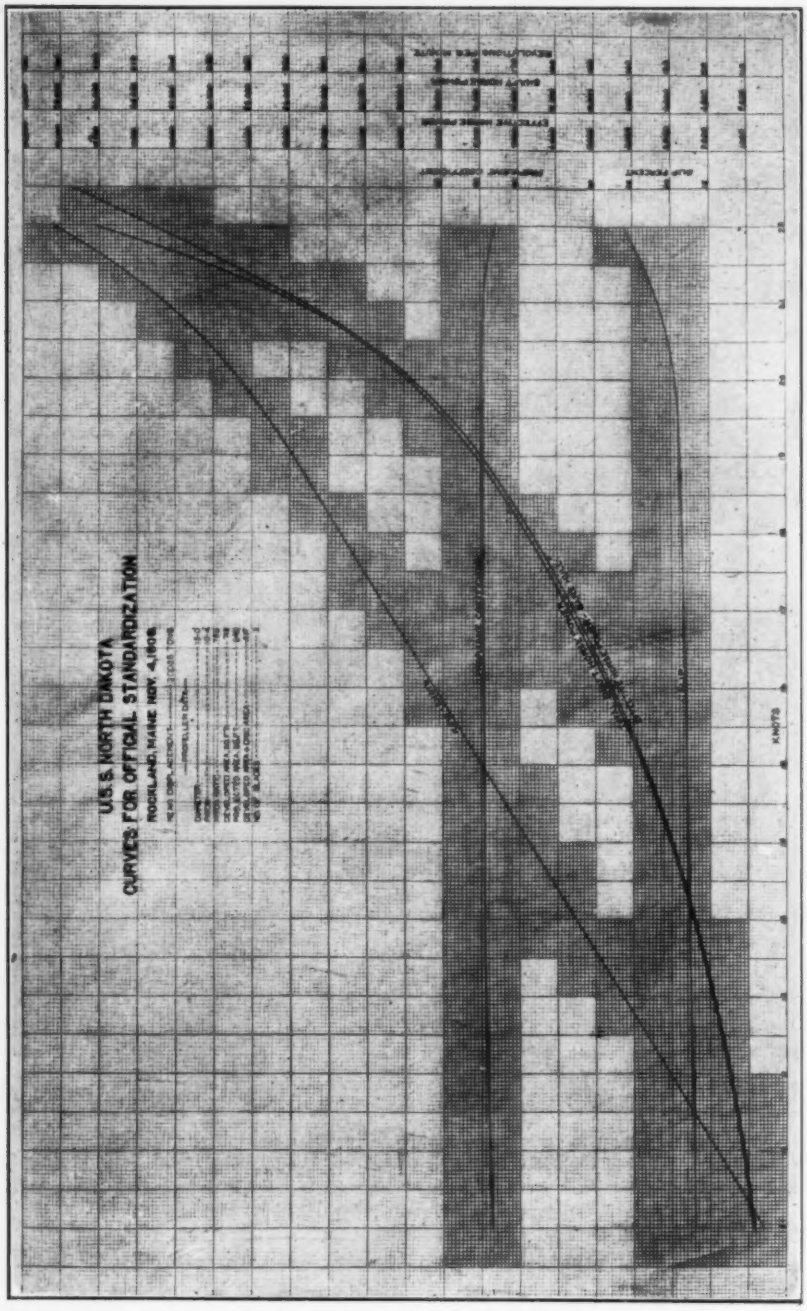


FIG. 15 HORSE POWER, SPEED AND REVOLUTION CURVES OBTAINED ON THE OFFICIAL TRIAL TRIP OF THE VESSEL

Note the rapid rise in power required as the speed increases.

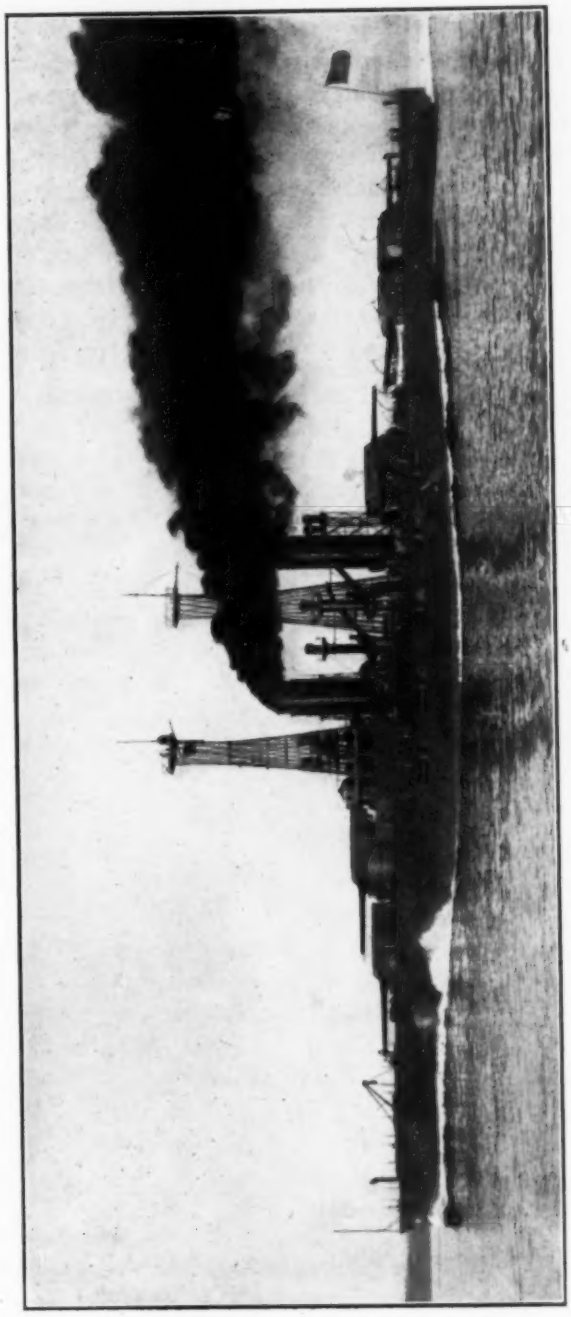


FIG. 16 THE NORTH DAKOTA STEAMING AT 22 KNOTS

This view, taken during the trial trip, shows the formation of bow and stern waves.

A number of additional views were shown and described, which are not reproduced, among which were some of the auxiliary machinery and a number of novel details of construction. These features include a refrigerating plant for cold storage purposes, ice-making, and cooling the magazine; the evaporating and distilling plant used to provide the vessel with fresh water, which has a capacity of 25,000 gallons in 24 hours, an engineering feature seldom found except on a battleship; a typical radiator, designed to heat the vessel in cold climates and to cool it in tropical regions, making comfort in any climate possible; the interior of the galley, with steam kettles and other means of cooking for the crew of 934 men, where a considerable amount of steam is used, constituting an engineering problem in itself.

A GRAPHICAL METHOD FOR CALCULATING STRESSES IN A CONNECTING ROD

By Winslow H. Herschel, New York

Member of the Society

Connecting rods are usually calculated by formul e. While empirical formulae are sufficiently accurate for slow engines where the inertia forces may be neglected, at how high a speed such formulae may be used is always a question. Thurston[1] gives two formulae, one for engines of moderate speed and the other for fast engines, but even in this case it is difficult to determine at what number of revolutions per minute we must discard one formula in favor of the other.

2 Analytical formulae are also unsatisfactory because they are based on incorrect assumptions. In the course of this paper I shall first enumerate the usual assumptions, and, taking up each in turn, show how it may be avoided, and finally give three numerical examples of the application of the new method.

3 *Assumption a.* The formula for calculating the bending moments due to the inertia forces may be written

$$M_1 = \frac{wR\omega^2 (l^2 x - x^3)}{6gl} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where

M_1 = bending moment due to the normal components of the inertia forces in in. lb.

l = distance between the crosshead and crank pins, or length of rod, in in.

x = distance from the crosshead pin to the section under consideration, in in.

R = crank length, in in.

[1] R. H. Thurston. A Manual of the Steam Engine, p. 103.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

g = acceleration of gravity, in in. per sec., squared

ω = angular velocity = $\frac{\pi n}{30}$, where n = r.p.m.

w = weight of an inch length of rod, in lb.

The definition of w indicates that the rod is of uniform section, although it is more common to use a tapered connecting rod for high speeds.

4 *Assumption b.* In deriving [1], Merriman[2] refers to a figure showing the crank at right angles with the connecting rod, that is, the piston has passed through about 44 per cent of the stroke. Unwin[3] shows the crank at right angles with the center line of the cylinder, that is, at 55 per cent stroke. Neither authority offers any proof of his assumption that the maximum bending moment occurs at the position indicated.

5 *Assumption c.* If $x = l\sqrt{\frac{1}{3}}$ in [1], M_1 will have a maximum value of $0.0638\frac{wR\omega^2 l}{g}$. Hence it is assumed that the maximum bending moment due to the inertia forces is at a section distant from the crosshead pin 0.6 (more accurately 0.577) the length of the rod.

6 *Assumption d.* Since the inertia forces are usually regarded as the sole cause of bending, the maximum section of the rod is often placed at 0.6, the length of the rod from the crosshead pin.

7 Marks[4] corrects [1] for horizontal engines, by introducing the bending moments due to the static weight of the rod, and thus finds

$$M = \frac{wR\,\omega^2\,(l^2\,x - x^3)}{6\,gl} + \frac{wlx}{2}\left(1 - \frac{x}{l}\right) \ldots\ldots\ldots [2]$$

The total bending moment, as given by [2], will have a maximum value when

$$x = l\left[-\frac{g}{R\omega^2} \pm \sqrt{\frac{1}{3} + \frac{g}{R\omega^2} + \left(\frac{g}{R\omega^2}\right)^2}\right]$$

This shows that the section of maximum bending moment will approach a point distant 0.577 the length of the rod from the crosshead end as the speed increases, and will approach the middle of the rod as the speed and consequently the inertia forces diminish.

[2] Mansfield Merriman. Mechanics of Materials, p. 257.
[3] W. C. Unwin. Elements of Machine Design, vol. 2, p. 105.
[4] W. D. Marks. The Relative Proportions of the Steam Engine, p. 160.
[5] L. L. Willard. Corliss Engine Design. The Engineer, Dec. 1, 1906, p. 774.

FOLDER No. 2

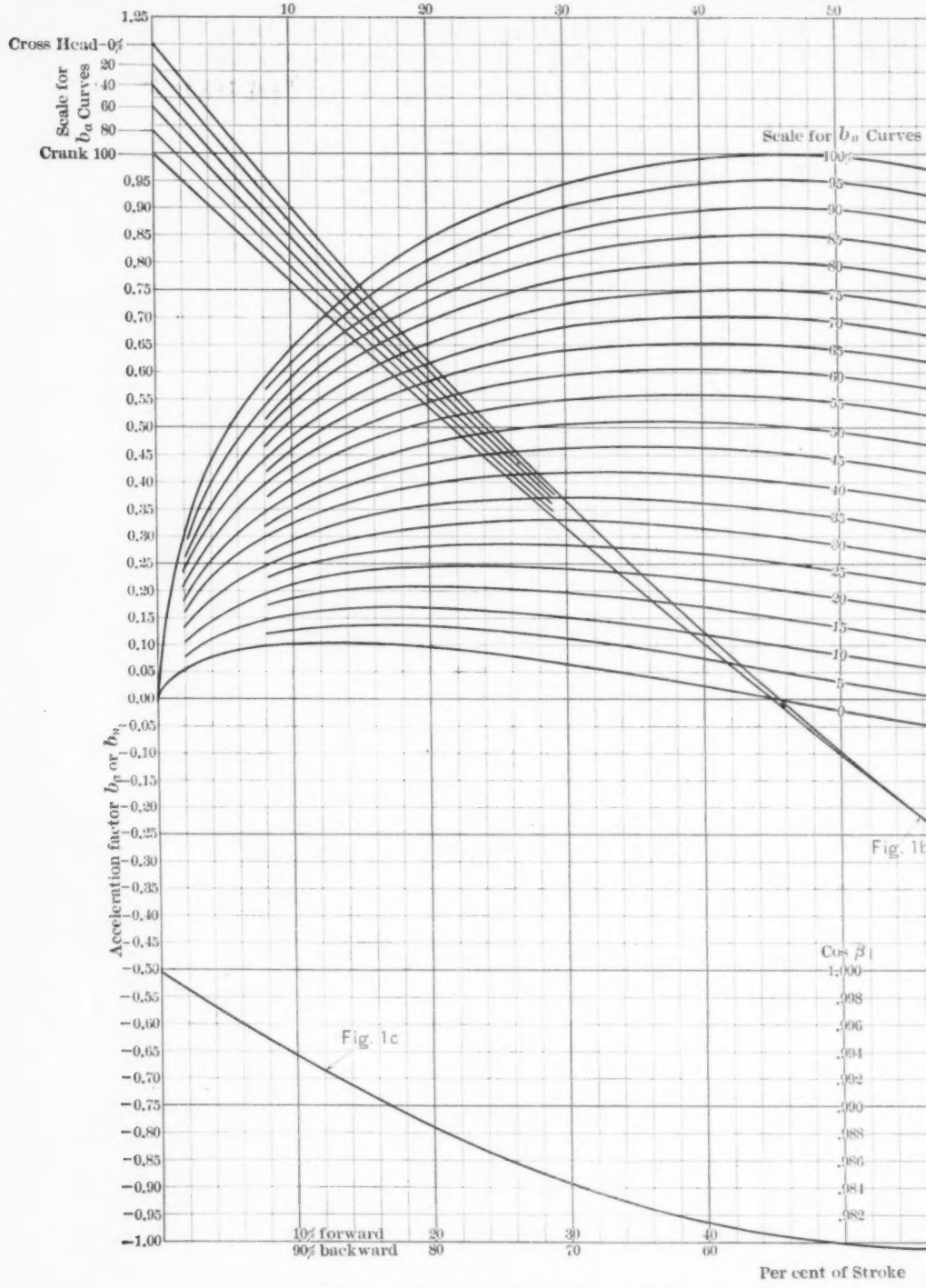
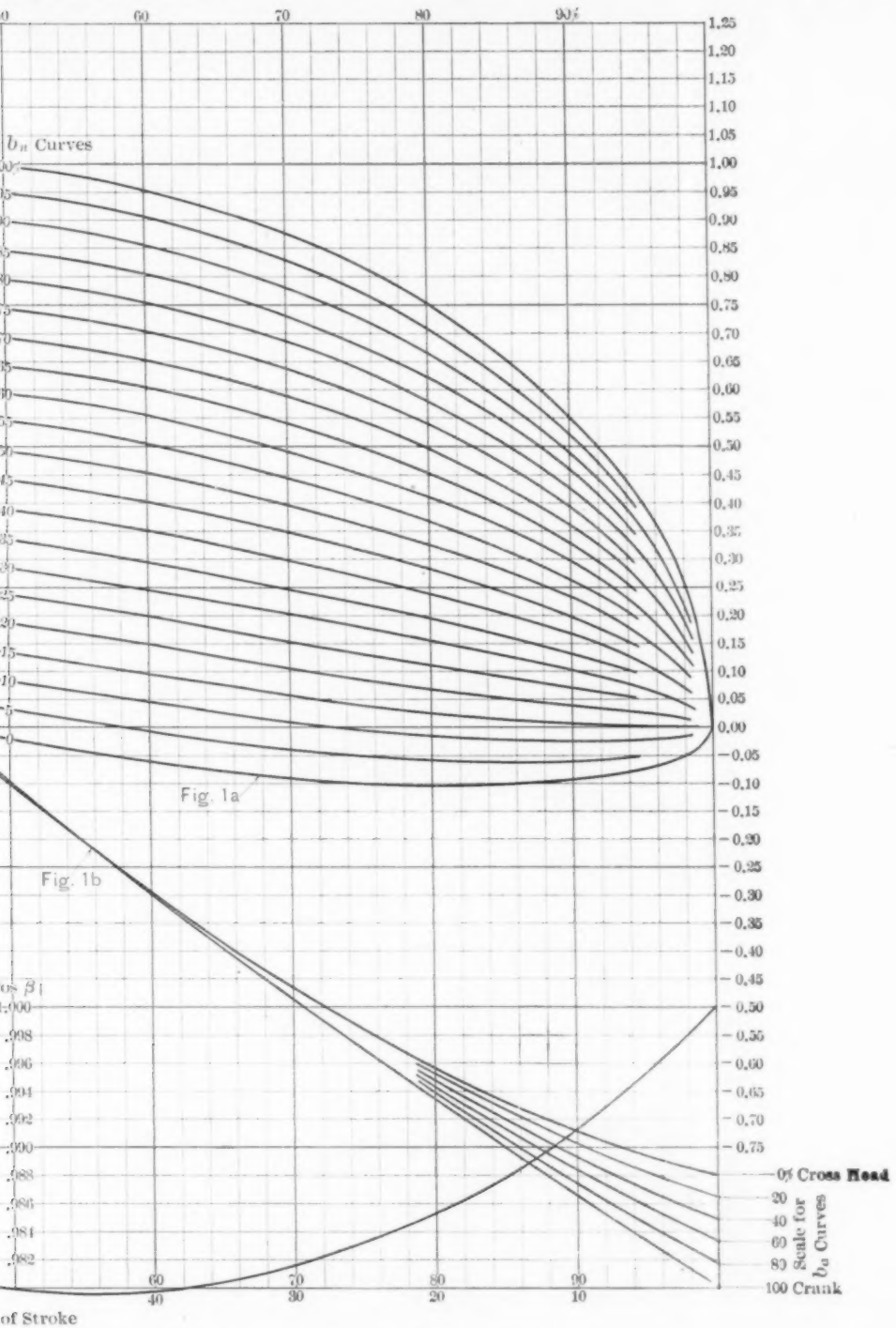


FIG. 1 DIAGRAM TO DETERMINE INERTIA FORCES IN CONNEC



of Stroke

IN CONNECTING ROD. CYLINDER NOT OFFSET. $l/r = 5$.

8 L. L. Willard[6] uses a graphical method which enables him to take account of the taper of the rod so far as it influences the bending moment due to the static weight. He assumes a rod of uniform section, however, in calculating the bending moment due to the inertia forces and the stress due to buckling and compression.

9 *Assumption e.* The axial thrust on the rod is taken as the piston area, multiplied by the unit pressure upon it and divided by cos β, where β is the angle between the connecting rod and the direction of stroke.

10 *Assumption f.* In addition to assuming, as in all structures, that the material is homogeneous, it is taken for granted in the case of a connecting rod that the workmanship is perfect, so that the rod may be considered as a column with no eccentricity of loading.

11 *Assumption g.* The maximum stress in the rod is taken as located at the section of maximum bending moment and as equal to the sum of the uniform stress due to the axial load, and the maximum stress due to the maximum bending moment. Expressed as a formula

$$\frac{P}{A} + \frac{M}{W} = K \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

where

P = axial load on rod, in lb.
A = section of rod, in sq. in.
M = bending moment, in in. lb.
$W = I/c$ = section modulus, in in. cubed
K = maximum stress in section, in lb. per sq. in.

12 If now we seek to avoid these seven assumptions, it soon becomes evident that we must use graphical methods. The analytical method would require a different formula for each shape of rod, and the calculations would be too complicated to be of practical value. I have therefore devised a graphical method, and shall show in the following paragraphs, that it obviates the necessity of making the seven assumptions.

DISCUSSION ON ASSUMPTIONS

13 In assumption *a*, since we wish to avoid the assumption that the rod is of uniform section, we must discard [1], in place of which we need some formula or other means of finding the normal component of the inertia force of a material point, which we will call P_n.

14 At the crank end of the rod the value of P_n will be

$$\frac{w}{g} R\omega^2 \sin(\alpha+\beta) = RwC \sin(\alpha+\beta)$$

where

w = weight of material point, in lb.
α = crank angle
β = rod angle
R = crank length, in in.
C = centrifugal force of 1 lb., 1 in. from center
(C can be obtained from a table or diagram.)

15 The inertia force at the crosshead end may be obtained from the formula

$$\text{inertia force} = \frac{bw}{g} = \frac{w}{g} R\omega^2 k = \text{approximately } \frac{wR}{g} \omega^2 (\cos\alpha + \lambda \cos 2\alpha)$$

where

b = acceleration, in in. per sec., per sec.
λ = ratio of crank to rod length

16 In a paper entitled Influence of the Connecting Rod upon Engine Forces, Sanford A. Moss[6] gives the more accurate formula

$$k = \cos\alpha + \frac{\left(\frac{1}{\lambda}\right) \cos 2\alpha + \sin^4 \alpha}{\left[\left(\frac{1}{\lambda}\right)^2 - \sin^2 \alpha\right]^{\frac{3}{2}}}$$

and calls attention to the typographical error in Unwin's Elements of Machine Design[7], of a minus instead of a plus sign in the numerator of the fraction.

17 In Hütt [8], is the formula

$$k = \frac{\cos(\alpha+\beta)}{\cos\beta} + \frac{\lambda \cos^2 \alpha}{\cos^3 \beta}$$

which agrees with the exact formula of Unwin. In all these formulæ, however, careful attention must be paid to the geometrical signs of the sines and cosines. There is less chance of error in the graphical method shown by Autenrieth[9]. If we introduce in place of $\frac{\omega^2}{g}$, ts

[6] Trans. Am. Soc. M. E., vol. 26, p. 375.

[7] W. C. Unwin. Elements of Machine Design, New edition, 1902, p. 117.

[8] Hütte des ingenieurs Taschenbuch, Akademischer Verein Hütte, Berlin, ed. 20, p. 829.

[9] Ed. Autenrieth. Beitrag zur Bestimmung der Trägheitskräfte einer Schubstange, in Zeitschrift des Vereins deutscher Ingenieure, vol. 39, 1895, p. 716.

equivalent C, we find that P_n at the crosshead end will be equal to $wCRk \sin \beta$.

18 We now have equations for the normal components of the inertia forces at both ends of the rod and may find intermediate values by simple interpolation If b_n is an inertia factor varying between $k \sin \beta$, and $\sin (\alpha + \beta)$, then P_n for any point in the rod may be determined by the formula

$$P_n = wCRb_n \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

19 Fig. 1a gives values of b_n for a normal crank mechanism where $\lambda = \frac{1}{5}$. The scale for b_n curves shows distance of material point from crosshead end in per cent length of rod. A plus sign on the out stroke or a minus sign on the back stroke before b_n shows that the force is acting upward in the case of a horizontal engine running over.

20 In cases where λ is not equal to $\frac{1}{5}$, a new diagram must be constructed in place of Fig. 1. $\lambda \sin \alpha = \sin \beta$, from which equation β may be calculated, and the percentage of stroke for any crank angle is equal to $50 \left[1 + \frac{1}{\lambda} - (\cos \alpha + \frac{1}{\lambda} \cos \beta) \right]$.

21 If the cylinder is offset, k may be found graphically, or the analytical method may be used as described by Professor Phetteplace in his paper, Offsetting Cylinders in Single-Acting Engines[10].

22 By means of Fig. 1a or a similar figure for the given conditions, and [4] we may find the value of P_n for each of the material points of which it is assumed the rod is composed. And since these forces produce neither a uniform nor a concentrated loading, we may most conveniently use the graphical or string polygon method for finding the bending moments M_1.

23 In assumption b, if we use Fig. 1a and [4] instead of [1] to determine M_1, it is evident that we are no longer restricted to the consideration of any particular part of the stroke, and we may find by trial at which point of the stroke M_1 attains its maximum value. Fig. 1a shows that b_n for the middle point of the rod, or the average value of b_n for all points of the rod, reaches a maximum at about 38 per cent stroke, and this may be taken as the approximate position for maximum bending moment.

24 In assumption c, the graphical method of finding M_1 locates the position as well as the value of the maximum bending moment due to the inertia forces. The error in assuming that the section of

[10] Trans., Am. Soc. M. E., vol. 31, p. 223.

maximum bending moment is located 0.6 the length of the rod from the crosshead end is not material, but it leads to the more serious error of placing the maximum section of the rod at this point.

25 In assumption *d*, in addition to the bending moments due to P_n, bending is caused by the static weight of the rod, as pointed out by Marks[4] and Willard[5], both of whom, however, use [1] to calculate M_i and assume a horizontal rod in calculating the bending moments due to the static weight, which we will call M_w. By using the string polygon method, with the rod and forces drawn in their true directions, and by allowing for the taper of the rod in calculating the weights of the material points, values of M_w may be found accurately and without difficulty. On this account and because with a vertical engine M_w becomes negligible, t will not need further consideration. If taken account of in the case of a horizontal engine, it should be added to M_i.

26 The main error in the usual method of calculation is in neglecting the bending moment due to the axial load. The total bending moment, M, is equal to $M_i + M_f$, where M_i is the bending moment due to the normal components of the inertia forces, and M_f is equal to $P\ (e + f)$, P being the axial load on the rod, e the eccentricity of loading, and f the deflection of the rod.

27 For the determination of f it is most convenient to use Mohr's graphical method, described in a paper by Willibald Trinks on The Deflection of Beams by Graphics.[11] This method of finding deflections consists, briefly stated, in using a mean corrected bending moment in place of a force in the string polygon method of finding a bending moment. The bending moments are corrected by multiplying by the ratio $\frac{I_o}{I}$, I_o being a constant moment of inertia, usually that of the largest section of the beam, and I being the moment of inertia of each section where the deflection is to be found.

28 The length from center to center of supports is divided into a number of equal parts and the mean corrected bending moment found for each part. These mean values are then laid off on a force scale and treated just as if they were forces acting through the centers of the parts into which the beam was divided. The work of drawing a string polygon is then completed, just as when finding bending moments by that method, the only difference being in the determination and significance of the pole distance.

[11] Trans., Am. Soc. M. E., vol. 24, p. 116.

29 Here we may use the formula

$$H = \frac{E I_o q}{lSxm} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [5]$$

where

H = pole distance
E = modulus of elasticity, in lb. per sq. in.
I_o = unit moment of inertia, in in. to the fourth power
q = number of equal parts into which beam is divided
l = length of beam between points of support, in in.
S = scale used in plotting corrected bending moments, being the number of in. lb. equal to 1 in.
m = number of times beam is reduced in size, in horizontal scale of string polygon, *e.g.*, if the scale is 3 in. = 1 ft., then $m = 4$
x = number of times deflection is to be enlarged on drawing.

q is selected according to the degree of accuracy required, and S is determined by the necessity of having the force scale a reasonable length. Then it is most convenient to calculate the value of $\frac{E I_o q}{lSm}$, after which x may be selected so as to bring the value of H as desired, that is, about half the length of the force scale, taking care that x is of such a value that the deflection may be easily scaled.

30 In assumption *e*, the axial thrust on the rod, P, is the sum of three quantities. The first is the pressure on the piston divided by $\cos \beta$ the second is the inertia force of the purely reciprocating parts, divided by $\cos \beta$, and the third is the axial component of the inertia forces of the rod itself, which we will call P_a.

31 Values of P_a are obtained by a method similar to that shown for finding P_n in Par. 22. At the crank end of the rod

$$P_a = \frac{w}{g} R\omega^2 \cos(\alpha + \beta) = RwC \cos(\alpha + \beta)$$

At the crosshead end of the rod

$$P_a = RwCk \cos \beta$$

or for any point on the rod

$$P_a = RwCb_a \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

Values of b_a are given on Fig. 1b, the scale for the b_a curves showing distance of material point from crosshead end, in per cent of length of rod. A plus sign on the out stroke or a minus sign on the back stroke

before b_a shows that the force is acting towards the crosshead. Fig. 2 shows the relative values and directions of the inertia forces at both ends of the connecting rod when at $\frac{1}{4}$ and $\frac{3}{4}$ stroke. Fig. 1 also contains a curve marked Fig. 1c giving values of cos β.

32 In assumption *f*, Moncrieff [12] in his paper on The Practical Column under Central or Eccentric Loads shows that an eccentricity of loading of only a few thousandths of an inch has a very marked influence on the bending moment. As a result of the investigation of numerous tests on columns, he concludes that these tests could be made to harmonize if an eccentricity of loading were assumed large enough

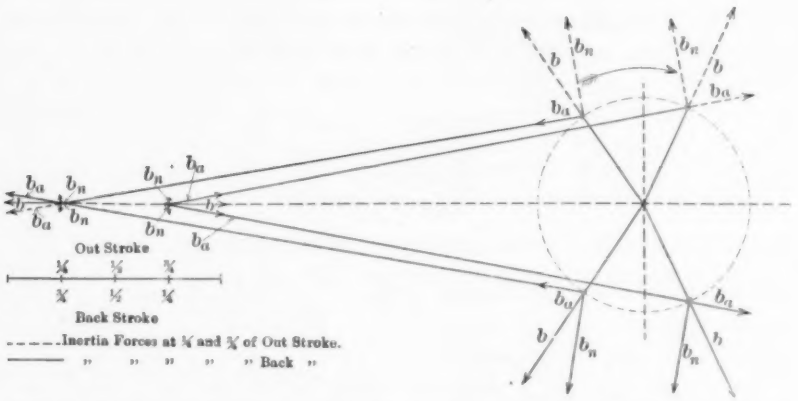


FIG. 2 RELATIVE VALUES AND DIRECTIONS OF INERTIA FORCES AT BOTH ENDS OF THE CONNECTING ROD WHEN AT $\frac{1}{4}$ AND $\frac{3}{4}$ STROKE

to cover the lack of homogeneity of the material as well as defects in alignment. He recommends taking a value of 0.6 for the ratio $\frac{ce}{r^2}$ where c is the distance to the extreme fibre, e is the eccentricity and r is the radius of gyration. This would give, for circular sections, the formula

$$e = 0.075\, d \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

Moncrieff does not consider tapered columns, but since [7] is at best only approximate, we may take d equal to the mean diameter of the rod.

33 In assumption *g*, in order to locate the section of maximum stress it is necessary to apply [3] to all sections of the rod. It would

[12] J. M. Moncrieff, Trans. Am. Soc. C. E., vol. 45, 1901, p. 334-445.

be possible first to find the deflections due to the bending moments M_1, add e, and take M_f equal to the product of their sum by P. From the total bending moments, $M = M_1 + M_f$, new deflections could be found, and this operation could be repeated until the deflections were those due to M, and no further increase in deflection could be obtained.

34 The same result may be obtained more quickly by using the formula for increase in deflection of a column due to the load P when the column has an initial maximum deflection, v, due to normal forces. The formula is

$$\Delta = \frac{Pl^2\left(e + \frac{5}{6}v\right)}{8\,EI - \frac{5}{6}Pl^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

where

Δ = increase in deflection due to load P
e = eccentricity of loading
v = maximum deflection due to side forces

This formula is derived by Mon rieff in his paper [2] and will give approximate results in our case if I is the moment of inertia of the section with mean diameter.

35 If v has been found by Mohr's method[11] and Δ and e have been calculated by [7] and [8], we may find the maximum value of M_f from the formula

$$M_f = P\,(e + f) = P\,(e + v + \Delta) \quad \ldots\ldots\ldots\ldots [9]$$

and the maximum bending moment to be resisted will be $M = M_1 + M_f$.

36 In cases where the rod is not of circular section it should be remembered that the bending moment in one direction is $M_1 + M_f$ and in the other direction simply M_f. In any case we may multiply the maximum bending moment by the ratio $\frac{I_o}{I}$ and assume the corrected bending moment thus obtained to be constant for the whole length of the rod, for if the maximum stress is constant, the corrected bending moment must be nearly so. Starting with this constant corrected bending moment we may proceed as before to find the true deflection by repetitions of Mohr's method.

37 For designing a rod of uniform stress with a circular section [3] may be brought into a more convenient form

$$\frac{\frac{P}{\pi d^2}}{4} + \frac{\frac{M}{\pi d^3}}{32} = K$$

$$M + \frac{Pd}{8} = \frac{K\pi d^3}{32}$$

or finally

$$d = \sqrt[3]{\frac{M + 0.125\,dP}{0.0982k}} \quad \text{.....[10]}$$

Equation [10] may be solved simply by assuming the value of d inside the cube root sign and recomputing when necessary.

PROPOSED METHOD OF DETERMINING STRESSES

38 We now have all the formulae needed to determine the stresses in an existing rod or to design a new rod with constant stress without need of the seven assumptions above stated, and I will next summarize the method to be employed in each case.

39 If we omit the ends of the rod, taking only the shaft, and divide it into equal parts of not too great length, we may assume without appreciable error that the center of gravity of each part is at the center of the part. This is of great convenience because it makes the lines of force coincide with the lines at the center of each of the strips used in finding deflections graphically. If the lines of force are taken through the actual centers of gravity of the material parts, it should be especially observed that the strips used in finding deflections must be equal, since otherwise [5] does not apply.

40 After finding the weight of each part of the rod, we must find values of P_n from [4] with the help of Fig. 1a. We find graphically the bending moments caused by the forces P_n and from these values of M_1 we find the deflections of the rod by Mohr's method[11], using [5] to find the pole distance. The maximum deflection thus found will be the value of v to be used in [8] together with the value of e from [7]. Equation 9 now gives the first trial value of the maximum bending moment due to buckling, and adding the maximum value of M_1 already obtained graphically and multiplying by $\frac{I_o}{I}$ for the middle of the rod, we have a trial value of the corrected bending moment, which is to be taken as constant for the whole length of the rod. Using this constant bending moment, we obtain new deflections by Mohr's method, but

these deflections are now values of f in [9] instead of their maximum being the value of v in [8]. We now proceed as before to find M_f, M and $\frac{MI_o}{I}$ and repeat the operations until no further increase in f may be obtained. The final values of M are then used in [3] to find K, the maximum stress at each section.

41 If we wish to design a rod that shall have a uniform stress, we may either start with a constant corrected bending moment or assume at first that the deflections are the same as found for an assumed rod. M is then found as before, but instead of multiplying by $\frac{I_o}{I}$ we must add $\frac{Pd}{8}$, divide by $0.0982\,K$ and extract the cube root in order to find new values of d by [10]. From these new values of d we get $\frac{I_o}{I}$ and then $\frac{MI_o}{I}$ and are ready to repeat Mohr's method to get new values of f and thus of M. When final values of M have been obtained by successive repetitions of the process, the value of K as obtained from [3] should be constant and equal to the value of K assumed in [10].

NUMERICAL EXAMPLES OF APPLICATION OF THE METHOD

42 The proposed method may become clearer from the three following numerical examples. We will assume a vertical gas engine with a 16 in. piston, 12 in. crank, 5 ft. rod, and a maximum explosion pressure of 400 lb. per sq. in. which may occur at any point of the stroke. As will appear later, the axial thrust in rod due to inertia forces is small, so that P is practically equal to the maximum pressure on piston, or say $P = 80{,}000$ lb.

43 Problem 1 is to find the stresses in an assumed rod of uniform taper, $3\frac{3}{8}$ in. in diameter at the crosshead end, and $3\frac{5}{8}$ in. in diameter at the crank end, neglecting the weight of the rod ends. The numerical work is shown in Table 1.

44 Fig. 3 shows the graphical work of all three problems, with the various operations numbered. At the right may be seen the force scales, with only one horizontal ray drawn to locate the pole. At the top, in the form of inverted arches, are shown a few corrected bending moment diagrams. These are not absolutely necessary, unless the lines of force do not coincide with the center lines of the

TABLE 1 NUMERICAL WORK FOR PROBLEM NO. 1

1 Distance of Section from Crosshead, in per cent Length of Rod.	2 Weight of Material Point in lb.	3 b_n for 38 per cent Stroke from Fig. 1a	4 $Pn = 17.26 \times$ Col. 2 $\times$ Col. 3	5 M_1 from Operation No. 1	6 Diameter of Rod = d	7 $I = \frac{\pi d^4}{64}$	8 $\frac{I_0}{I}$	9 $\frac{M_1 I_0}{I}$	10 Deflection from Operation No. 2	11 Deflection from Operation No. 3	12 $f + e =$ Col. 11 + 0.262	13 $M_f =$ Col. 12 $\times$ 80,000	14 $M =$ Col. 13 + Col. 5	15 $\frac{MI_0}{I} =$ Col. 14 $\times$ Col. 8
0		0.033			3.37	6.33	1.320				0.262	210	210	277
5	15.2	0.081	21.2	1500	3.39	6.48	1.288	1930	0.0030	0.019	0.281	225	240	309
15	15.5	0.176	47.2	4400	3.41	6.64	1.258	5530	0.0088	0.050	0.312	250	294	369
25	15.7	0.272	73.7	7080	3.44	6.87	1.215	8600	0.0137	0.069	0.331	265	336	408
35	15.9	0.368	101.2	9280	3.46	7.03	1.188	11030	0.0173	0.081	0.343	274	367	436
45	16.2	0.464	129.	10890	3.49	7.28	1.147	12480	0.0194	0.088	0.350	280	389	446
55	16.4	0.559	159.	11740	3.51	7.45	1.120	13160	0.0198	0.088	0.350	280	397	444
65	16.6	0.655	188.	11580	3.54	7.71	1.082	12500	0.0181	0.081	0.343	274	390	422
75	16.9	0.751	218.	10320	3.56	7.88	1.059	10920	0.0147	0.069	0.331	265	368	390
85	17.1	0.846	249.	7710	3.59	8.15	1.025	7880	0.0096	0.050	0.312	250	327	335
95	17.3	0.942	281.	3310	3.61	8.34	1.000	3310	0.0034	0.019	0.281	225	258	258
100		0.990			3.62	8.43	0.990				0.262	210	210	208

1 Distance of Section from Crosshead, in per cent Length of Rod	16 Deflection from Operation No. 4	17 $f + e =$ Col. 16 + 0.262	18 $M_f =$ Col. 17 $\times$ 80,000	19 $M =$ Col. 18 + Col. 5	20 $\frac{MI_0}{I} =$ Col. 19 $\times$ Col. 8	21 Deflection from Operation No. 5	22 $f + e =$ Col. 21 + 0.262	23 $M_f =$ Col. 22 $\times$ 80,000	24 $M =$ Col. 23 + Col. 5	25 $A = \frac{\pi d^2}{4}$	26 $\frac{P}{A} = \frac{80,000}{\text{Col. } 25}$	27 $W = \frac{\pi d^3}{32}$	28 $\frac{M}{W} = \frac{\text{Col. } 24}{\text{Col. } 27}$	29 $K =$ Col. 20 + Col. 28
0									210	8.92	8990	3.76	5580	14570
5	0.014	0.276	221	236	304	0.014	0.276	221	236	9.02	8880	3.82	6180	15060
15	0.037	0.299	239	283	355	0.036	0.298	239	283	9.13	8770	3.89	7280	16050
25	0.056	0.318	254	325	394	0.055	0.317	254	325	9.29	8620	4.00	8130	16750
35	0.069	0.331	265	358	425	0.067	0.329	263	356	9.40	8520	4.07	8750	17270
45	0.075	0.337	270	379	434	0.073	0.335	268	377	9.57	8370	4.17	9030	17400
55	0.075	0.337	270	387	433	0.073	0.335	268	385	9.68	8270	4.24	9070	17340
65	0.068	0.330	264	380	411	0.066	0.328	262	378	9.84	8130	4.35	8700	16830
75	0.055	0.317	254	357	378	0.054	0.316	253	356	9.95	8040	4.43	8050	16090
85	0.037	0.299	239	316	324	0.036	0.298	239	316	10.12	7910	4.54	6970	14880
95	0.013	0.275	220	253	253	0.013	0.275	220	253	10.23	7820	4.62	5480	13300
100									210	10.29	7770	4.66	4510	12280

strips used in finding deflections, but it is convenient to plot them in order readily to see the changes of corrected bending moment with successive repetitions of Mohr's method. At the bottom of Fig. 3 is drawn the rod as assumed for problem 1, and as calculated in problem 2. The main part of the figure is covered with string polygons used in finding M_b, v, and f.

45 With $R = 12$ in. and a speed of 225 r.p.m., $C = 1.4386$, and by [4], $P_n = 17.26\ wb_n$, values of P_n being given in column 4 of Table 1.

46 Graphical operation No. 1 gives the bending moments due to inertia forces, by the simple string polygon method, using the loads given in column 4. From the assumed diameters of the rod we must next get the ratios $\frac{I_o}{I}$ in order to correct these bending moments by Mohr's method,[11] in graphical operation No. 2, from which we get the deflections of co umn 10. The maximum value, 0.0198 in., is equal to v, and we may now find by [8]

$$\Delta = \frac{80{,}000 \times (60)^2 \left[0.075 \times 3.5 + \left(\frac{5}{6} \times 0.0198 \right) \right]}{(8 \times 30{,}000{,}000 \times 7.37) - \frac{5}{6} \times 80{,}000 \times (60)^2} = 0.0525 \text{ in.}$$

By [7]

$$e = 0.075 \times 3.5 = 0.2625 \text{ in.}$$

and thus by [9]

$$M_f = 80{,}000\ (0.2625 + 0.0198 + 0.0525) = 26{,}800 \text{ in. lb.}$$

and adding the maximum value of M_1 from column 5 we get $M = 38{,}540$. But for the mean diameter of 3.5 in.

$$\frac{I_o}{I} = \frac{8.34}{7.37} = 1.133$$

so that our corrected bending moment is equal to

$$38{,}540 \times 1.133 = 43{,}700$$

47 This value is assumed constant, that is, the corrected bending moment diagram would be a straight line if shown in Fig. 3. Thus the distances laid off on the force scale, in operation No. 3, are equals and give deflections as shown in column 11. Adding the value of e, already found, we get $f + e$ of column 12, from which M_f, in column 13, is found by [9]. Column 13 and all subsequent bending moments in Table 1 are expressed in hundreds of in. lb.

48 Having now both M_i and M_f, their sum multiplied by the ratio $\frac{I_o}{I}$ gives the new corrected bending moments of column 15, and these are used in operation No. 4 to obtain new deflections. Repeating in columns 17 to 20 the same calculations as in columns 12 to 15 we again return to our graphical method in operation No. 5 to

TABLE 2 NUMERICAL WORK FOR PROBLEM NO. 2

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Distance of Section from Cross-Head, in per cent Length of Rod	Deflection from Operation No. 6	$f+e$ = Col. 2 + 0.262	M_f = Col. 3 × 400,000	M = Col. 4 + Col. 5 of Table 1	Pd = 50,000 × Col. 6 of Table 1	Col. 5 + Col. 6	d^3 = Col. 7 / 3930	$d = \sqrt[3]{\text{Col. 8}}$	I	$\frac{o}{I}$	MI_o/I = Col. 11 × Col. 5	Deflection from Operation No. 7	$f+e$ = Col. 13 + 0.378	M_f = Col. 14 × 400,000	M = Col. 15 + Col. 5 of Table 1
0		0.262	105	105	169	274	69.6	4.11	14.0	3.03	318		0.378	151	151
5	0.167	0.429	172	173	170	343	87.2	4.43	18.9	2.25	389	0.031	0.409	164	165
15	0.425	0.687	275	279	171	450	114.6	4.85	27.2	1.565	437	0.081	0.459	184	191
25	0.630	0.892	357	364	172	536	136.3	5.14	34.3	1.239	451	0.120	0.498	199	208
35	0.764	1.026	410	419	173	592	150.7	5.32	39.3	1.080	453	0.146	0.524	210	221
45	0.830	1.092	437	448	174	622	158.0	5.41	42.0	1.010	453	0.159	0.537	215	227
55	0.830	1.092	437	449	175	624	158.9	5.42	42.4	1.000	449	0.159	0.537	215	227
65	0.764	1.026	410	422	177	599	152.5	5.34	39.9	1.064	449	0.145	0.523	209	219
75	0.630	0.892	357	367	178	545	138.8	5.18	35.3	1.202	442	0.120	0.500	200	208
85	0.425	0.687	275	283	179	462	117.6	4.90	28.3	1.500	424	0.081	0.459	184	192
95	0.167	0.429	172	175	180	355	90.2	4.48	19.8	2.15	376	0.031	0.409	164	167
100		0.262	105	105	181	286	72.7	4.18	15.0	2.84	298		0.378	151	151

1	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Distance of Section from Cross-Head, in per cent Length of Rod	$\frac{Pd}{8}$ = 50,000 × Col. 9	Col. 16 + Col. 17	d^3 = Col. 18 / 3930	$d = \sqrt[3]{\text{Col. 19}}$	I	$\frac{I_o}{I}$	$\frac{MI_o}{I}$ = Col. 16 × Col. 22	Deflection from Operation No. 8	$f+e$ = Col. 24 + 0.365	M_f = Col. 25 × 400,000	M = Col. 26 + Col. 5 of Table 1	$\frac{Pd}{8}$ = 50,000 × Col. 20	Col. 27 + Col.	d^3 = Col. 29 / 3930	$d = \sqrt[3]{\text{Col. 30}}$
0	205	356	90.3	4.48	19.8	1.586	239		0.365	146	146	224	370	94.0	4.55
5	222	387	98.3	4.61	22.2	1.415	233	0.022	0.387	155	156	230	386	98.2	4.61
15	243	434	110.7	4.80	26.1	1.203	230	0.057	0.422	169	173	240	413	102.5	4.68
25	257	465	118.3	4.91	28.5	1.102	229	0.083	0.448	179	186	245	431	109.5	4.78
35	266	487	124.0	4.99	30.4	1.032	228	0.101	0.466	186	195	250	445	113.2	4.83
45	270	497	126.6	5.02	31.2	1.008	229	0.110	0.475	190	201	251	452	115.0	4.86
55	271	498	126.8	5.03	31.4	1.000	227	0.110	0.475	190	202	251	453	115.2	4.86
65	267	486	123.7	4.98	30.2	1.040	228	0.100	0.465	186	198	249	447	113.7	4.85
75	259	467	119.0	4.92	28.8	1.090	227	0.083	0.448	179	189	246	435	110.8	4.80
85	245	437	111.2	4.81	26.3	1.195	229	0.057	0.422	169	177	240	417	106.0	4.73
95	224	391	99.3	4.63	22.6	1.390	232	0.022	0.387	155	158	231	389	99.0	4.63
100	209	360	91.5	4.50	20.1	1.563	236		0.365	146	146	225	371	94.3	4.55

TABLE 2—Continued

1	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47
Distance of Section from Cross-Head, in per cent Length of Rod	I	$\frac{I_o}{I}$	$\frac{MI_o}{I}$ = Col. 27 × Col. 33	Deflection from Operation No. 9	A	Weight of Material Point in lb.	b_n for 38 per cent Stroke, from Col. 3 of Table 1	b_n for 44 per cent Stroke, from Fig. 1a	P_n = 17.26 × Col. 37 × Col. 38	P_n = 17.26 × Col. 37 × Col. 39	M_1 from Operation No. 10	M_1 from Operation No. 11	$f + e$ = Col. 35 + 0.357	M_f = Col. 44 × 400,000	M = Col. 45 + Col. 42	$\frac{Pd}{8}$ = 50,000 × Col. 31
0	21.0	1.306	191		16.26								0.357	143	143	227
5	22.2	1.235	193	0.022	16.69	28.4	0.081	0.05	39	24	2900	2800	0.379	152	155	230
15	23.5	1.167	202	0.057	17.20	29.2	0.176	0.15	88	75	8200	8000	0.414	166	174	234
25	25.6	1.070	199	0.084	17.94	30.5	0.272	0.25	143	132	13200	12700	0.441	176	189	239
35	26.7	1.027	200	0.102	18.32	31.2	0.368	0.35	198	188	17200	16900	0.459	184	201	241
45	27.4	1.000	201	0.111	18.55	31.6	0.464	0.45	253	245	20100	19600	0.468	187	207	243
55	27.4	1.000	202	0.111	18.55	31.6	0.559	0.55	305	299	21400	21000	0.468	187	208	243
65	27.2	1.008	199	0.102	18.47	31.4	0.655	0.65	355	352	20900	20500	0.459	184	205	242
75	26.1	1.050	198	0.084	18.10	30.8	0.751	0.75	398	398	18200	17900	0.441	176	194	240
85	24.6	1.113	197	0.057	17.57	29.9	0.846	0.85	436	438	13100	12900	0.414	166	179	237
95	22.6	1.212	192	0.022	16.84	28.7	0.942	0.95	466	470	5300	5200	0.379	152	157	232
100	21.0	1.306	191		16.26			1.00					0.357	143	143	227

1	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62
Distance of Section from Cross-Head, in per cent Length of Rod	Col. 46 + Col. 47	d^3 = Col. 48 / 3930	d = $\sqrt[3]{\text{Col. 49}}$	I	$\frac{I_o}{I}$	$\frac{MI_o}{I}$ = Col. 46 × Col. 52	Deflection from Operation No. 12	$f + e$ = Col. 54 + 0.357	M_f = Col. 55 × 400,000	M = Col. 56 + Col. 42	A	P = 400,000 / Col. 58	W	M = Col. 57 / Col. 60	K = Col. 59 + Col. 61
0	370	94.2	4.55	21.0	1.296	185		0.357	143	143	16.26	246	9.25	155	40100
5	385	97.8	4.61	22.2	1.225	190	0.022	0.379	152	155	16.69	240	9.62	161	40100
15	408	103.9	4.70	23.9	1.139	198	0.057	0.414	166	174	17.35	231	10.19	171	40200
25	428	109.0	4.78	25.6	1.062	201	0.085	0.442	177	190	17.94	223	10.72	177	40000
35	442	112.2	4.82	26.5	1.027	206	0.103	0.460	184	201	18.25	219	10.99	183	40200
45	450	114.5	4.85	27.2	1.000	207	0.112	0.469	188	208	18.47	217	11.20	186	40300
55	451	114.5	4.85	27.2	1.000	208	0.112	0.469	188	209	18.47	217	11.20	187	40400
65	447	113.8	4.84	26.9	1.012	207	0.103	0.460	184	205	18.40	217	11.13	184	40100
75	434	110.5	4.80	26.1	1.042	202	0.085	0.442	177	195	18.10	221	10.86	180	40100
85	416	105.7	4.73	24.6	1.106	198	0.057	0.414	166	179	17.57	228	10.39	172	40000
95	389	99.0	4.63	22.6	1.203	189	0.022	0.379	152	157	16.84	238	9.74	161	39900
100	370	94.0	4.55	21.6	1.296	185		0.357	143	143	16.26	246	9.25	155	40100

get deflections of column 21. Similarly columns 22 to 24 give corrected values of columns 17 to 19.

49 It is now seen that further repetition would not materially

change the result, and taking the values of M in column 24 as final, we may proceed to find K by [3]. It is seen that the greatest stress is at 45 per cent the length of the rod from the crosshead end, not at 60 per cent as usually assumed.

50 Problem No. 2 is to design a rod for the same engine as in problem No. 1, so that there shall be a uniform stress of 40,000 lb. per sq. in. under a load of 400,000 lb., that is, with P five times as great as previously assumed. For a starting point we may take M_1, v and e from problem No. 1. Then by [8]

$$\Delta = \frac{400{,}000 \times 3600 \times 0.279}{(30{,}000{,}000 \times 8 \times 7.37) - \left(\frac{5}{6} \times 400{,}000 \times 3600\right)} = 0.707 \text{ in.}$$

This high value of Δ would indicate that our assumed rod is too small, since, as a rough rule, a column may be considered as no longer serving its purpose when the deflection is greater than $\frac{l}{100}$ which, in our case, would be equal to 0.6 in. By [9]

$$M_f = 400{,}000\ (0.2625 + 0.0198 + 0.707) = 396{,}000 \text{ in. lb.}$$

To this we add M_1, giving $M = 407{,}740$, and from this, multiplying by 1.133, the corrected bending moment is equal to 462,000.

51 In problem No. 1, graphical operation No. 2, we had by [5]

$$H = \frac{30{,}000{,}000 \times 8.34 \times 10}{60 \times 10{,}000 \times 200 \times 4} = 5.21 \text{ in.}$$

In operations Nos. 3 to 5 we kept this value for H by changing S from 10,000 to 40,000 and x from 200 to 50. Similarly, we may now in problem No. 2 change S to 400,000 in. lb. = 1 in. and change x to 5, and we shall still have $H = 5.21$ in.

52 By operation No. 6, using the constant corrected bending moment of 462,000 in. lb., we get the deflections as given in column 2 of Table 2. Taking the same value of e as in problem No. 1, we get $f + e$ in column 3, which multiplied by P gives values of M_f in column 4. Values M_f, M, $\frac{Pd}{8}$, $M + \frac{Pd}{8}$ and $\frac{MI_o}{I}$ in Table 2, are expressed in thousands of inch pounds. We then add values of M_1 and get values of M in column 5.

53 Taking values for d inside the cube root sign, in [10], from column 6 of Table 1, we get new values of d as shown in column

FOLDER No. 3

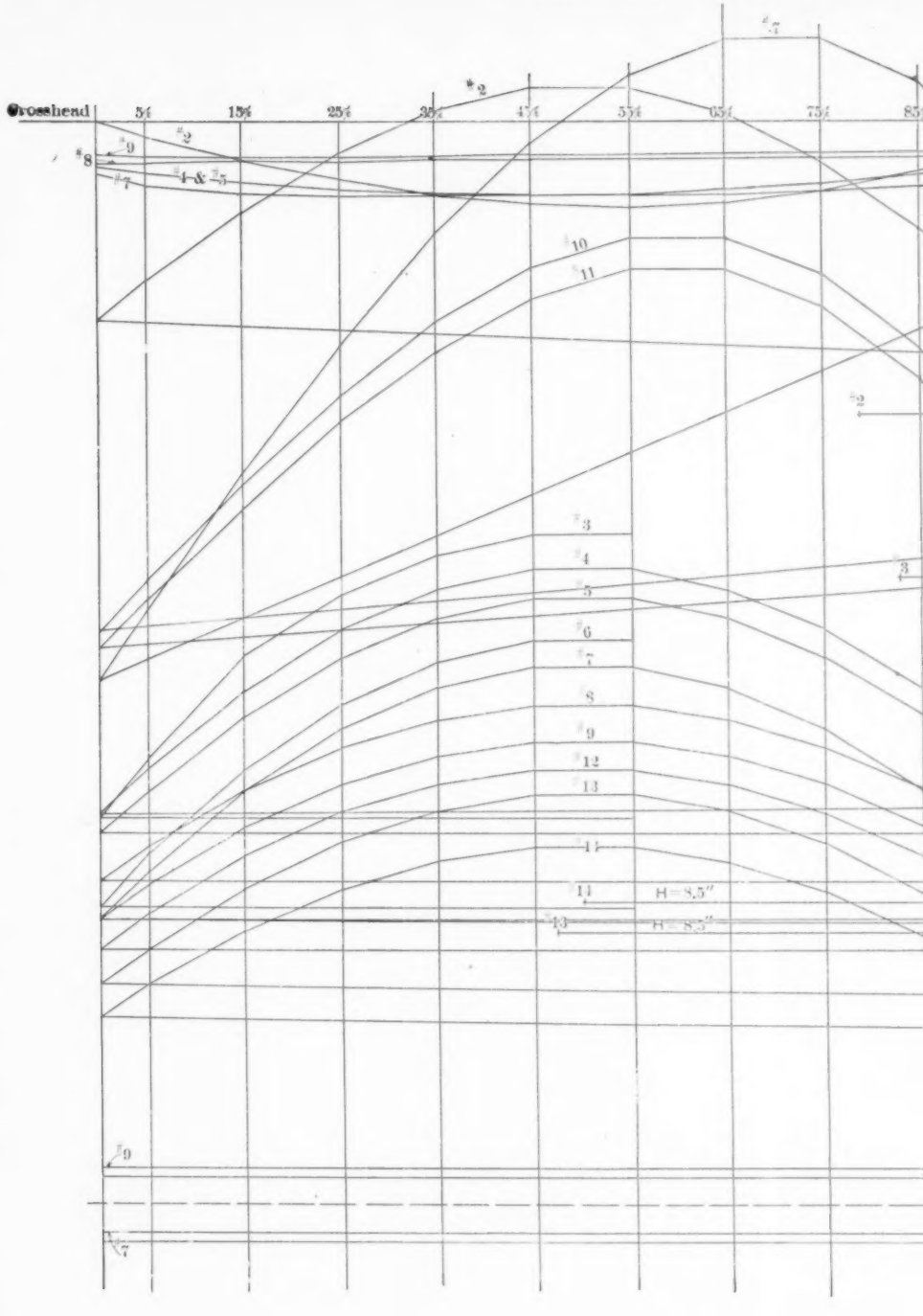
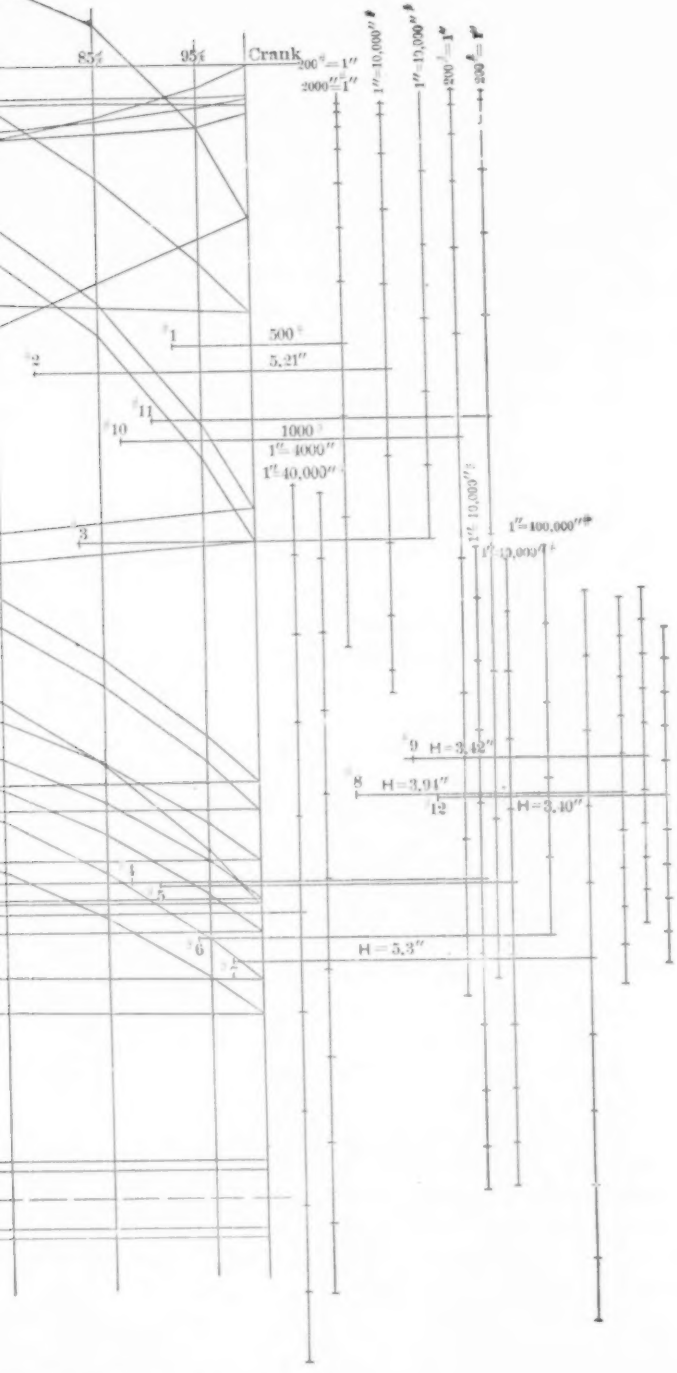


FIG. 3 GRAPHICAL WORK OF PROBLEMS 1,



ROBLEMS 1, 2 AND 3

UOF

9, and from these, new values of I, $\frac{I_o}{I}$ and $\frac{MI_o}{I}$. We are now ready to get, by operation No. 7, the new deflections shown in column 13. In operations Nos. 6, 7, 8, 9 and 12, values of H are proportiona to I_o and are respectively 5.21, 5.30, 3.94, 3.42 and 3.40. In this respect problem No. 2 is unlike problem No. 1 where H is constant. It should also be noticed that e is variable, since the value of d in [7] changes with each repetition of Mohr's method.

54 Columns 14 to 23 carry out the same computation as column 3 to 12 and, after obtaining by operation No. 8 the deflections of column 24, we repeat in columns 25 to 34 the same computation. Operation No. 9 now shows deflections so nearly the same as those previously obtained that further repetition of the same line of calculations seems unnecessary.

55 We have thus far constantly used values of M_i from column 5 of Table 1, and must now eliminate this source of error by finding new values derived from the weight of a rod with diameters as given in column 31. Weights thus obtained are given in column 37.

56 We have assumed that the maximum bending moment due to inertia forces will be caused when the piston has traveled 38 per cent of the stroke, while according to the usual assumption the piston would have traveled about 44 per cent. Reading from Fig. 1 values of b_n for 44 per cent stroke and taking values for 38 per cent stroke as already found, we find values of P_n from [3] in columns 40 and 41. This gives us finally, by operations Nos. 10 and 11, values of M_i at 38 per cent stroke in column 42, and at 44 per cent stroke in column 43. It will be seen that while the differences are not large, values in column 42 are somewhat larger than those in column 43, and we may therefore continue to assume that the maximum bending moment occurs at 38 per cent stroke. The exact position for the maximum bending moment will evidently depend upon the distribution of weight in the rod.

57 We are now ready to run through our computation in columns 44 to 53, similar to columns 25 to 34, using the new values of M_i of column 42 to add to M_f in column 45.

58 Graphical operation No. 12 gives deflections of column 54 and these are so nearly the same as those in column 35 that we may take them as final. Adding e, multiplying by P and adding M_i, we get our final bending moments in column 57. From our final diameters in column 50 we get A and W, and then K may be found by [3].

The stresses in columns 59 and 61 are expressed in hundredths of pound per sq. in., and their sum, in column 62, should be 40,000. This does not, however, serve as a check on the graphical work.

59 For a comparison with the usual methods we may calculate the maximum diameter of a connecting rod, for the conditions of problem No. 2, by the formulae given in Kent.[13] Since $P = 80{,}000$ lb.

$$D\sqrt{p} = \sqrt{\frac{4P}{\pi}} = 319$$

Using these values and that of $l = 5$ ft. we may obtain the following table

No. of formula . . .	3	5	6	7	9	10
Diam. of rod, in. . .	5.80	5.70	3.81	6.48 and 3.94	8.47	6.48 and 6.82

Since neither factors of safety nor maximum stresses are stated with these formulae it is difficult to make a choice between them.

60 In the German handbook Hütte[14] may be found the formula

$$P = \frac{\pi^2 EI}{S l^2}$$

where S is the factor of safety against buckling, for which a value of 25 is given, and E is the modulus of elasticity. Taking $S = 25$ and $E = 30{,}000{,}000$, the formula reduces in English units, for a rod of circular section, to

$$d^4 = \frac{Pl^2}{592{,}000}$$

where

d = maximum diameter of rod at center, in in.
P = axial thrust on rod, in lb.
l = length of rod, in in.

This gives for our case, $d = 4.70$ in. which happens to be the average of the diameter at the end and at the middle as found by the graphical method.

61 The maximum stress in this rod is however much less, as found by the usual analytical method than as found graphically. Using the formula given in paragraph 5, assumption *c*,

[13] Wm. Kent. Mechanical Engineers Pocket Book, p. 800.

[14] Hütte des ingenieurs Taschenbuch, Akademischer Verien Hütte, Berlin, ed. 20, p. 838.

$$M_1 \text{ maximum} = 0.0638 \frac{wR\omega^2 l^2}{g}$$

we find the maximum bending moment to be 19,550 in. lb. Assuming that the rod has a diameter of 4.70 in. at the point of maximum bending moment, the maximum stress, according to [3] would be

$$\frac{400{,}000}{17.40} + \frac{19{,}550}{10.20} = 24{,}900 \text{ lb. per sq. in.}$$

62 That the stresses found by the graphical method, and given in column 62 of Table 2, are so much greater is mainly due to the introduction of an eccentricity of loading. If $e = 0$, then, assum ng the deflections to remain as given n column 54

$$M_f = 400{,}000 \times 0.112 = 44{,}800 \text{ in. lb.}$$

Adding $M_1 = 21{,}400$ from column 42, we get $M = 66{,}200$ from which the maximum stress due to bending would be $\frac{66{,}200}{11.20} = 5920$ lb. per sq. in. which would be a great reduction from the value o 18,700 given in column 61. If we omit M_f entirely, M becomes equal to M_1 and the maximum stress due to bending is equal to $\frac{21{,}400}{11.20} =$ 1910 lb. per sq. in. or practically the same as found analytically.

63 Problem No. 3 is to find the stresses in the rod calculated in problem No. 2, when $P = 80{,}000$ lb. The method is similar to that used in problem No. 1, but we may simplify it somewhat by starting with the deflections as found for problem No. 2, instead of calculating Δ from v as before. The numerical work is shown in Table 3, in which all bending moments are expressed in hundreds of inch pounds. The graphical work is shown 'n Fig. 3, operat'ons Nos. 13 and 14.

64 Columns 17 and 18 of Table 3 are used in checking our assumption that $P = 80{,}000$ lb. We have assumed that the piston has made 38 per cent stroke, and that the pressure upon it is 80,424 lb. Then in column 17 we have values of b_a from Fig 1b and from [6], the axial component of the inertia forces is equal to

$$P_a = 17.26\, wb_a$$

Taking weights from column 37 of Table 2, we get values of P_a in column 18, and the sum of this column is 810.5 lb. The inertia force of the piston weighing 700 lb. will be

$$700 \times 17.26 \times 0.168 = 2030 \text{ lb.}$$

acting against the pressure on the piston. Then P will be equal to

$$\frac{80{,}424 - 2030}{0.982} - 810 = 79{,}100 \text{ lb.}$$

where we have assumed the value to be 80,000.

65 It should be noticed that, in the case of a column or connecting rod, the maximum stress is not directly proportional to the load

TABLE 3 NUMERICAL WORK FOR PROBLEM NO. 3

1 Distance of Section From Cross-Head, in per cent Length of Rod	2 M_f = Col. 44 of Table 2 × 80,000	3 M = Col. 2 + Col. 42 of Table 2	4 $\frac{Ml_0}{I}$ = Col. 3 × Col. 52 of Table 2	5 Deflection From Operation No. 13	6 $f + e$ = Col. 5 + 0.357	7 M_f = Col. 6 × 80,000	8 M = Col. 7 + Col. 42 of Table 2	9 $\frac{Ml_0}{I}$ = Col. 8 × Col. 52 of Table 2	10 Deflection From Operation No. 14	11 $f + e$ = Col. 10 + 0.357	12 M .11 × 80,000	13 M = Col. 12 + Col. 42 of Table 2	14 $\frac{P}{A}$ = 80,000 + Col. 58 of Table 2	15 $\frac{M}{W}$ = Col. 13 + Col. 60 of Table 2	16 K = Col. 14 + Col. 15	17 b_a for 38 per cent Stroke from Fig. 1b	18 P_a = 17.26 × Col. 17 × Col. 37 of Table 2
0	286	286	370		0.357	286	286	370		0.357	286	286	4920	3090	8010	0.168	
5	304	333	408	0.006	0.363	291	320	392	0.005	0.362	290	319	4790	3320	8110	0.166	81.3
15	332	414	472	0.015	0.372	298	380	433	0.013	0.370	296	378	4610	3710	8320	0.164	82.5
25	354	486	515	0.023	0.380	304	436	462	0.020	0.377	302	434	4460	4060	8520	0.161	84.7
35	368	540	554	0.028	0.385	308	480	492	0.025	0.382	306	478	4380	4340	8720	0.159	85.5
45	376	577	577	0.030	0.387	310	511	511	0.027	0.384	307	508	4330	4540	8870	0.156	85.0
55	376	590	590	0.031	0.388	311	525	525	0.027	0.384	307	521	4330	4640	8970	0.154	83.9
65	368	577	584	0.028	0.385	308	517	523	0.025	0.382	306	515	4350	4630	8980	0.151	81.8
75	354	536	558	0.023	0.380	304	486	505	0.022	0.379	303	485	4420	4470	8890	0.149	79.2
85	332	463	510	0.016	0.373	298	429	474	0.014	0.371	297	428	4560	4120	8680	0.146	75.3
95	304	357	429	0.006	0.363	291	344	413	0.005	0.362	290	343	4750	3520	8270	0.144	71.3
100	286	286	370		0.357	286	286	370		0.357	286	286	4930	3090	8010	0.143	

P. Thus if we assume in problem No. 2 that the ult mate streng h of the material is 80,000 lb. per sq. in., we can not say that the factor of safety is 10 although a stress of 40,000 lb. per sq. in. would be obtained with a load five times the actual maximum load. In problem No. 3 the maximum stress was found to be 8980, and hence the factor of safety would be

$$\frac{80{,}000}{8980} = 8.90$$

66 It has been assumed 'n the problems that the rod ends had no weight, or what is the same thing, that their inertia forces acted

directly on the pins and so brought no stress upon the shaft of the rod. That there may be considerable error in this assumption is pointed out by Bach[15] in his conclusion that the resultant of all the inertia forces of the rod, including the ends, may pass through the crank pin, or in he case of a rod with a very heavy crank end, may fall beyond the pin.

67 This assumption is, however, not a necessary part of the method here proposed, since all equations, with the exception of [7] and [10], apply to a rod of any section. If we wish we may calculate the necessary diameters of a rod of full length to include the ends, and then make the end sections of the same area and sect on modulus.

68 The method of calculation described in this paper furnishes a means of find ng the stress or the factor of safety at any point of a connecting rod. The usual assumptions are avoided, the only assumption necessary being in regard to the proper value of e to allow for lack of homogeneity of material as well as actual eccentric ty o loading. It is evident that the greater the eccentricity assumed, the smaller will be the factor of safety required.

[15] C. Bach. Die Maschinen-Elemente, p. 774.

INTERCHANGEABLE INVOLUTE GEARING

By Wilfred Lewis, Philadelphia, Pa.

Member of the Society

This paper was prepared at the request of the Meetings Committee of The American Society of Mechanical Engineers as the contribution to the discussion upon the need of a standard or standards for gears to be considered at the Joint Meeting with the Institution of Mechanical Engineers in July 1910. By way of explanation, it may be said that its author is chairman of a committee on Standards for Involute Gears, appointed about a year ago by the president of The American Society of Mechanical Engineers to investigate the subject of interchangeable involute gearing, and to recommend, if found desirable, a standard or standards. This paper, however, is not to be considered as an expression of the opinions of the other members of the committee, except where so stated.

2 After more or less unsatisfactory experience with cycloidal gearing, I investigated about twenty-five years ago the subject of involute gearing with the object of determining upon a system, for the firm of Wm. Sellers & Company, with which I was then connected. The conditions imposed called for a system applicable to any number of teeth between a 12-tooth pinion and a rack, without change in the Sellers addendum which had always been made 0.3-pitch for the cycloidal teeth, hitherto used almost exclusively by them.

3 I found that the involute forms then in vogue were confined to obliquities of $14\frac{1}{2}$ deg. and 15 deg. with an addendum equal to the modulus, or about 0.32-pitch. This long addendum with such small obliquities naturally gave rise to interference between racks and pinions of less than 30 teeth, and rather than modify the involute form I finally recommended the adoption of a pressure angle of 20 deg. At the same time, I was well aware of the fact that even this obliquity was not sufficient to prevent interference between a 12-

Presented at the Birmingham, England, Meeting (July 1910) of The American Society of Mechanical Engineers and The Institution of Mechanical Engineers.

toothed pinion and a rack, but for such pinions and the gears of 60 teeth or less, with which they commonly engaged, I believed the interference would not be noticeable in practice. I was strongly tempted to go further and fix upon an obliquity of $22\frac{1}{2}$ deg., but 20 deg. then appeared such a radical departure from common usage that the advantages of the greater angle were dismissed as being possibly more theoretical than real.

4 The 20-deg. system with an addendum of 0.3-pitch has now been in use by Wm. Sellers & Company for twenty-five years and has given satisfaction in a general way, although the interference referred to has been more or less noticeable on 12-toothed pinions. I reviewed this matter ten years ago in a paper read before the Engineers' Club of Philadelphia, advocating an obliquity of $22\frac{1}{2}$ deg. and suggesting as a much-needed reform in engineering practice the consideration of uniformity in interchangeable gearing. I then pointed to the action of the Franklin Institute more than thirty years earlier, which inaugurated a standard system of screw threads and expressed the hope that by the interchange of opinions an agreement among engineers might be reached leading to the gradual disappearance of needless diversity in the forms of gear teeth.

5 Nothing in this direction had been done however, when the subject of interchangeable involute gear-tooth systems was brought to the attention of The American Society of Mechanical Engineers in a paper by Ralph E. Flanders presented in December 1908. A number of systems in general use were analyzed and their merits discussed from various points of view and the desire expressed that the Council be petitioned to appoint a committee to investigate the subject of interchangeable involute gearing and, if found desirable, to recommend a standard or standards.

6 In answer to this petition the Council voted in January 1909, that the President appoint a committee of five members to formulate standards for involute gears and present the same to the Council. Without anticipating in any way the conclusions of this committee yet to be formulated, if indeed an agreement be possible, I believe it will be helpful to give publicity to the line of investigation upon which we have embarked and thus obtain the benefit of such criticism or encouragement as it may provoke.

7 At the first meeting of the committee in June 1909, it was decided to obtain an expression of opinion from the manufacturers of gears and gear cutters and later in October the following circular letter was sent out:

Dear Sir:

A committee of The American Society of Mechanical Engineers has been appointed to consider the subject of standards for interchangeable involute gears, and would like to have as much light upon the subject as can be given them by the manufacturers.

Without limiting in any way the type of gears to be used for any special service, it is believed to be desirable that some type should be known as standard so that any gears of that standard may run together and be perfectly interchangeable.

The committee is, therefore, interested to know the extent to which involute gears are modified in your practice to avoid interference, and it would be pleased to have your ideas as to what, in your opinion, should constitute standard interchangeable involute gears. Any suggestion you may be pleased to offer will be very much appreciated.

Yours truly,

WILFRED LEWIS, *Chairman*
HUGO BILGRAM
E. R. FELLOWS
C. R. GABRIEL
GAETANO LANZA

Committee on Standards for Involute Gears

8 In response to the circular letter above referred to, about one hundred answers were received, expressing more or less interest in the subject and giving more or less conflicting preferences and conclusions.

9 One correspondent said he would hail with delight any system whereby complete interchangeability could be attained on gears running without noise up to 1000 ft. a minute, as among automobile manufacturers noiseless gearing was always the chief end in view.

10 It is undoubtedly true, as pointed out by other correspondents, that perfect cutters do not necessarily produce perfect gears and that close attention must be paid to the setting of the cutter, proper indexing, speed, feed, etc., and from the extent to which the present $14\frac{1}{2}$-deg. system is established, it has been argued that even if the pressure angle of $14\frac{1}{2}$-deg. were not the most desirable, it would be better for the sake of the established practice to let it alone. Quite a number share this opinion and argue against the recommendation of a standard which may simply add another system to those already in use.

11 It may be said in reply that the chief advantages of involute gearing over cycloidal gearing which it has pretty generally displaced are the comparative simplicity of the involute curve and a slight variation in center distances not permissible in any other system of gearing. Despite the difficulty of securing general recognition for

any system of interchangeable gearing, I believe the committee should investigate and report upon some ideal standard or standards.

12 Attention is called by a number of correspondents to the increasing number of gears cut by the hobbing process and the difficulty of making such gears interchangeable with certain cut gears, thus indicating the hobbing principle as the first principle to which all cutters should conform, the rack being the basis of all generated gears. It is admitted that hobbed gears can be used interchangeably with gears cut by certain other processes, but some doubt is expressed of the possibility of realizing an interchangeable system using pinions with a small number of teeth and teeth of reasonable length without resorting to some modification of the involute curve. The opinion is freely expressed that if such modification becomes necessary the method should be known and clearly defined, so that makers of gears can adopt it and, if necessary, make their own cutters.

13 As pointed out in Mr. Flanders' paper and as mentioned repeatedly by our correspondents, the most desirable quality in gearing and the one by which it is almost universally judged, is quietness and smoothness of running. Next to this come strength, durability and permanence of form, and upon the last, of course, depend continued quietness and smoothness of action. Friction and journal pressure are of less importance, but still worth considering, and before reaching any conclusions from theoretical considerations alone, we propose to determine if possible, in a practical way, the relative advantages of some of the systems in common use and, with these, other systems to which we are disposed to give favorable consideration.

14 It is understood that the Institution would be pleased to have the coöperation of the Society in the work they have in hand looking to a determination of the friction in the transmission of power by gearing. In this connection we have been reminded of a suggestion of Prof. J. Burkitt Webb of Stevens Institute of Technology made soon after the publication of the Sellers' experiments upon gears reported to our Society in 1885. In these experiments an attempt was made to measure the friction loss between the teeth of a pair of spur gears, but the apparatus used, made originally for testing the friction of worm gearing, was not delicate enough for the purpose and the errors introduced exceeded, in some cases, the net result.

15 Professor Webb then suggested the possibility of so dividing one of the pair of spur gears to be tested as to make the load on the

teeth self-contained. The apparatus which we have designed embodies this idea, thus making it possible to run gears under heavy loads at high speeds with a very small consumption of power. We have also provided in our apparatus for an adjustment of center distance and means to measure the thrust between centers while the gears are running. Of course, the thrust between centers can be estimated very closely for involute gears from the pressure angle on the teeth, but we anticipate results somewhat in excess of this on account of the excess in friction of approach over that of recess, and, if any but involute gears are tested, it will also be interesting to compute from experimental data the effective obliquities of other systems.

16 We propose to determine the friction loss under various speeds and pressures for wheels and pinions cut to the Brown and Sharpe $14\frac{1}{2}$-deg. standard, the 20-deg. stub tooth and a $22\frac{1}{2}$-deg. tooth with addendum of $\frac{7}{8}$ module or about 0.278 pitch. These gears will be tested at normal center distance, and also at distances about 1 per cent or 2 per cent of the pitch greater or less than this, and an effort will be made to record graphically the noise produced under these different conditions.

17 We believe that accuracy and permanence of form can thus be given their proper influence on the reduction of noise. It may take some time to determine the effect of wear, but from the method of loading the teeth and the small amount of power consumed some indication of the tendency of wear can be obtained. All gears tend to wear out of shape, and involute gears more so than cycloidal, but we recognize as a possibility that this tendency may be checked by the deformation itself and also that the loss in friction at different parts of a gear tooth is practically incalculable on account of the variations in friction for different velocities of sliding.

18 The experiments we propose should therefore give information unobtainable in any other way and throw a flood of light on the problem in hand.

19 As it is quite impossible for any of the committee, who are all busy men, to make the experiments here outlined, we have been fortunate enough to interest, through the intervention of Professor Lanza of the Massachusetts Institute of Technology, two of his undergraduates, Messrs. Green and Doble, in making these experiments the subject matter for a graduating thesis. Professor Lanza hoped to have the work completed in June so that the results might be communicated for discussion at our Joint Meeting in July but this has proved impossible, partly because of the magnitude of the under-

taking and partly of the delay in the completion of the testing machine.

20 The apparatus used in making these experiments is shown by the photographs, Fig. 1 and Fig. 2, and the line drawing, Fig. 3, which gives some of the principal dimensions and shows the knife edges on which the machine rests. The machine consists of a frame A designed to carry a pinion shaft in roller bearings at one end, and a frame B pivoted to it and designed to carry the gear wheels W engaging with a wide-faced pinion P on the pinion shaft. The frame B is held to the

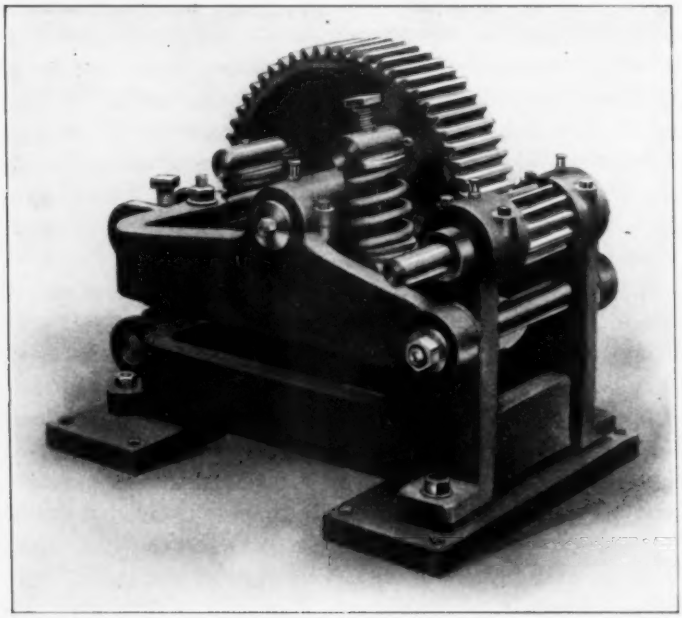


FIG. 1 GEAR TESTING MACHINE

frame A at its outer end by an adjustable clamping bolt C, and provision is made to measure the thrust on centers by means of the spring D acting between the frame A and an adjustable abutment on the frame B. The gear wheels to be tested consist of a central gear with a wide face and two side gears with narrow faces. The central gear carries two heavy cross pins G, which pass through clearance holes in the side gears, and the side gears carry two heavy pins H, which pass through clearance holes in the central gear. Between the projecting

ends of these pins G and H, heavy helical springs S are inserted, upon which pressure can be applied by means of the set screws K.

21 The pressure of these four springs S is resisted by the gear teeth, the middle gear pressing against one side of the pinion teeth and the side gears pressing against the other side. The pinion thus becomes simultaneously a driver and a driven gear and the power required to turn it when loaded in this way is only that required to overcome the friction of the teeth and whatever resistance there may be in the gear journals. The latter presumably is very small

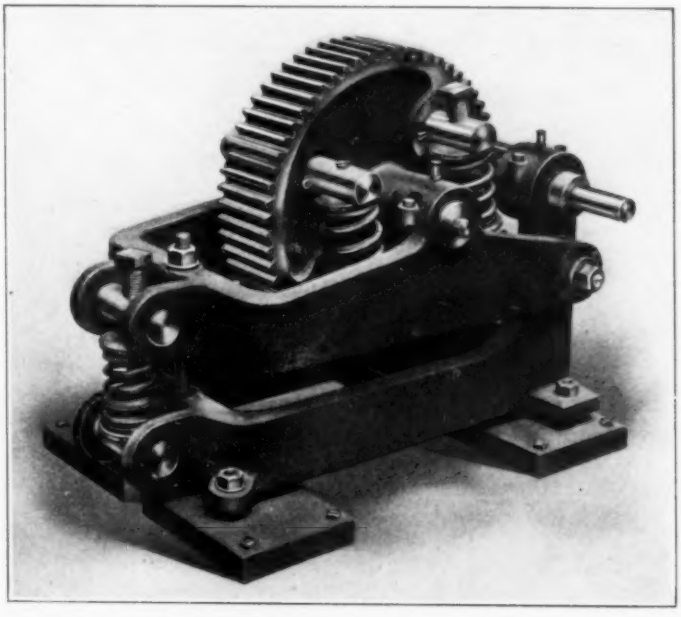


FIG. 2 VIEW SHOWING ARRANGEMENT OF PIVOTED FRAME AND SPRING FOR MEASURING THRUST

indeed, but provision has been made to measure it by substituting plain cylinders without teeth for the gears and pinion, and running these under the same journal pressures. By deducting the resistance due to journals from the total resistance with running gears, the friction of the teeth alone can be determined.

22 In operation this machine is driven by an extension to the pinion shaft, carried to bearings several feet distant to permit of ample flexibility. The knife edge directly beneath the pinion rests

upon a permanent support, and the other knife edge is carried upon the platform of a small platform scales. The driving moment in the pinion shaft will therefore be measured by the scale reading at the end of an arm 20 in. long, and by reversing the direction of motion given to the pinion shaft the effect of any initial lateral strain in the driving shaft can be eliminated.

23 Fig. 4 illustrates three types of gearing to be tested by Messrs. Green and Doble, and with these other types may be included later on.

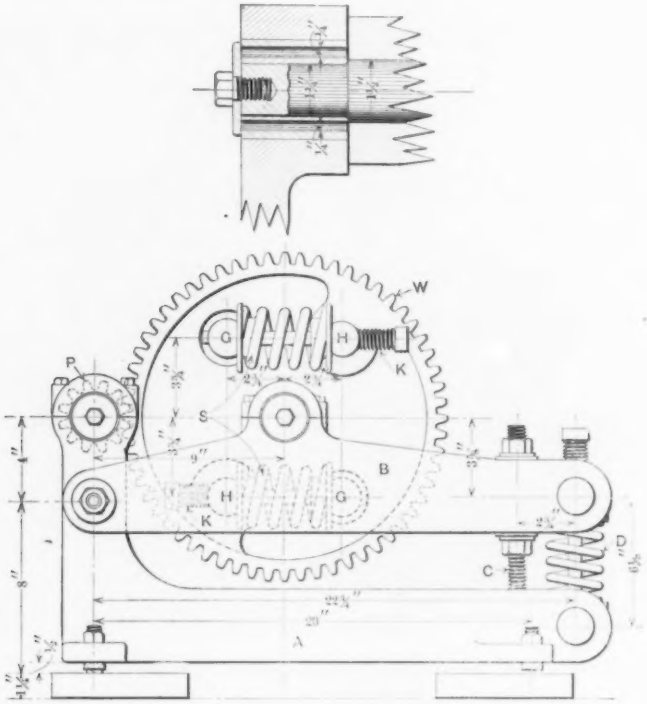


FIG. 3 DIAGRAM OF GEAR TESTING MACHINE

24 Figs. 5, 6 and 7 illustrate a group of involute gears designed by Mr. Bilgram to engage a rack of 15-deg. obliquity, and to demonstrate the possibility of using pinions of ten or even nine teeth with such a rack, provided the addendum can be varied. Without wishing to advocate the use of a variable addendum in interchangeable gearing, it is interesting to note the possibility of making a tentative solution of the problem in this way. A set of these gears has kindly been

furnished by Mr. Bilgram for making a comparative test. He has also made a set of models, from which the figures have been photographed, and referring to them, he gives the following explanation:

25 "While the involute system of gearing has decided advantages over any other, it has the one disadvantage that the faces of the

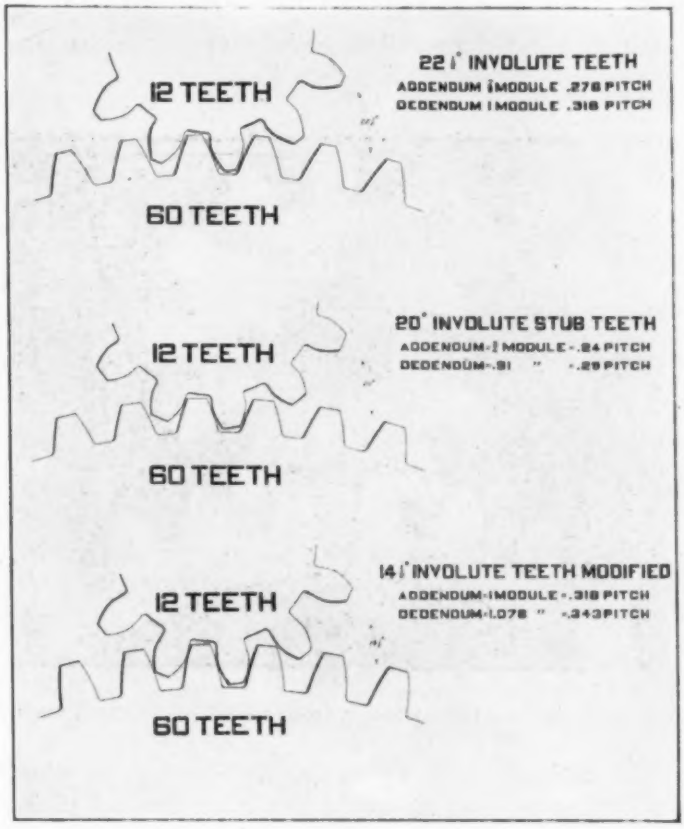


FIG 4 GEAR TYPES TESTED IN THE MACHINE

teeth of wheels come into interference with the flanks of pinions, if the latter have a comparatively small number of teeth. Unless the flanks of the latter are undercut, the teeth will interlock or at least mesh improperly.

26 "In making a single pair of wheels, a remedy can readily be applied. There are two ways in which interference can be avoided,

namely either by increasing the angle of pressure or by shortening the addendum of the wheel. If the latter method is chosen and it is desired not to reduce the working depth of the teeth, it is necessary to add to the addendum of the pinion the amount taken from the addendum of the wheel.

27 "This latter method is out of the question when the problem is given to make an interchangeable set of spur wheels from a rack down to a 12-tooth pinion. This problem may be solved by a combination of both remedies alluded to."

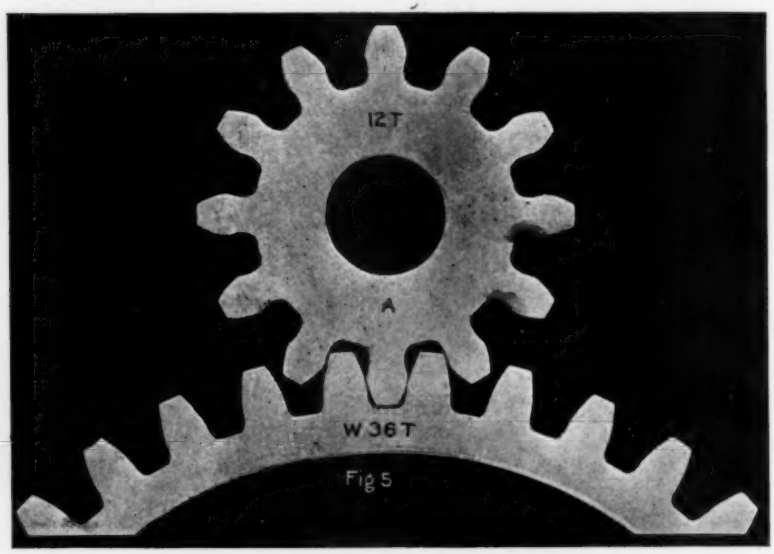


FIG. 5 BILGRAM SYSTEM: PINION WITH UNDERCUT TEETH

28 The method consists of making racks and larger wheels with normal addendum, but increasing the addendum of pinions just enough to prevent the rack tooth from interfering with the flank. The samples presented (Figs. 5 to 7) consist of a rack R and a 36-tooth wheel W, with angle of pressure of 15 deg. and addendum equal to the modulus. The 12-tooth pinion A, generated by a rack, corresponding to rack R, shows the undercutting thereby produced. Obviously this pinion will not work, as so much of the involute is cut away that the path of contact is materially less than one pitch. But there are also shown pinions of twelve, ten and nine teeth, made with increased addenda. These were generated by a rack like R,

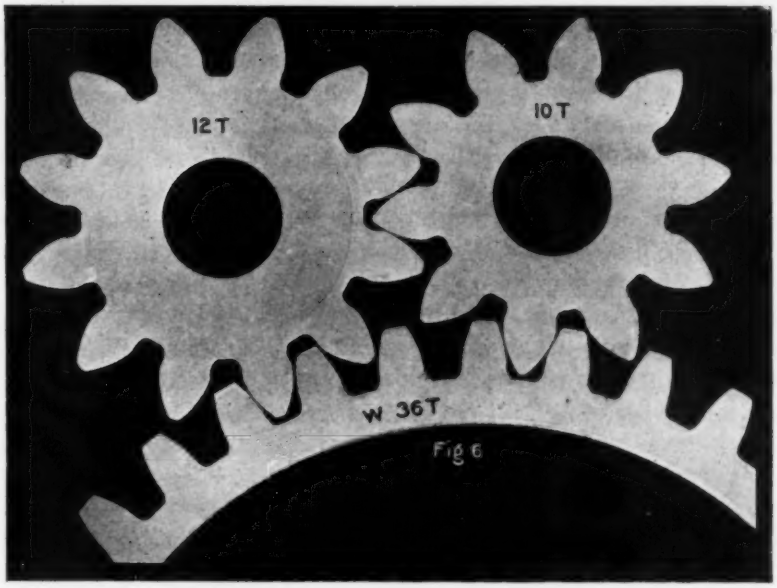


FIG. 6 BILGRAM SYSTEM: PINIONS WITH INCREASED ADDENDA IN MESH WITH GEAR

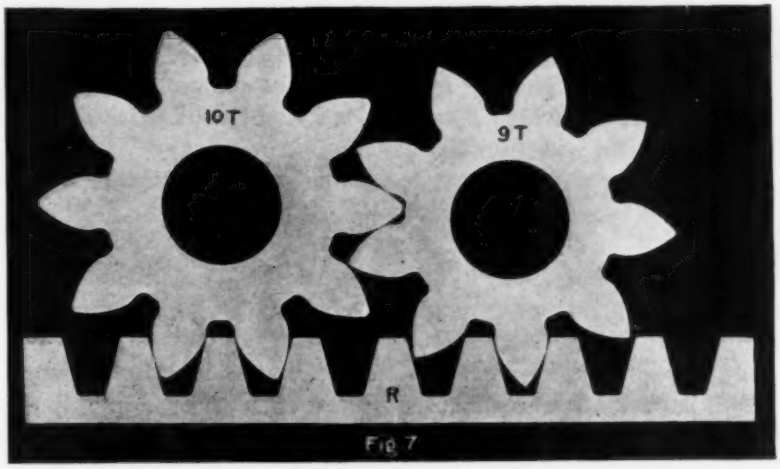


FIG. 7 BILGRAM SYSTEM: PINIONS WITH INCREASED ADDENDA IN MESH WITH RACK

but with a somewhat greater addendum than that used in generating the wheel *W* and a somewhat greater cutting depth. If these pinions are then mated with wheels of a large number of teeth, they will not enter as far as with pinions of equal number of teeth and have therefore a slightly less working depth.

29 On this plan may be based a system of involute gearing with a working depth of twice the modulus, and with a moderate pressure angle, 15 deg. in the samples submitted. Pinions of a small number of teeth will have an increased addendum, but a theoretically correct action is maintained. The pinion teeth have a wide base and are strong. One disadvantage in those cases in which pinions with less than about 24 teeth are embraced, is that the center distance is greater than that computed by the usual rule from the modulus and the number of teeth of the meshing wheels. Moreover, pinions will not have the full working depth when meshing with large wheels or with racks, but even in the case of a 10-tooth pinion meshing with a rack, the path of contact exceeds one pitch so that at least for a portion of the action two teeth will be in contact simultaneously.

30 The plan proposed by Mr. Fellows is to use an involute with an angle of pressure of 20 deg. and to reduce the addendum to $\frac{3}{4}$ of the modulus. Such teeth are known as "stub teeth". By this method interference in case of a rack gearing with a 12-tooth pinion is just avoided and in the case of two 12-tooth pinions meshing with each other the path of contact is equal to about $1\frac{1}{8}$ of the pitch.

31 Mr. Gabriel prefers the $14\frac{1}{2}$-deg. standard of the Brown and Sharpe Manufacturing Company and he has prepared a contribution to the discussion expressing his views on the subject.

32 The system which I propose is that of a pressure angle of $22\frac{1}{2}$ deg. and an addendum of $\frac{7}{8}$ of the modulus.

33 I believe that an interchangeable system of involute gearing, to be of the greatest value, should extend from a 12-tooth pinion to a rack, and in the selection of gears to be tested we have chosen a 12-tooth pinion engaging a 60-tooth wheel. The maximum reduction with the maximum strength in a limited space is the problem in gearing that generally confronts the engineer and a ratio of five to one is very often as much as he can realize without sacrificing too much strength. I recognize, of course, that the adoption of a larger number of teeth in the smallest allowable pinion overcomes some difficulties, and that this may be a debatable point, but I do not think any system of interchangeable gearing will be satisfactory which does not include pinions of twelve teeth. The cycloidal

system formerly employed was based upon a 12-tooth pinion, and I believe this number can be retained for the smallest number in an interchangeable involute system without serious objection. That is, there is better ground for the retention of this minimum number than for a higher number and when the pros and cons have all been summed up there will be no change in this well-established minimum for interchangeable pinions.

34 Although the experiments made under the direction of Professor Lanza are not by any means conclusive, enough has been done to indicate that the friction loss in gear teeth is influenced to a greater extent by the length of the addendum than by the obliquity of the system. Theoretically, the friction loss in involute teeth is independent of the obliquity and increases with the addendum. The loss in journal friction should vary as the secant of the pressure angle, but the latter is also affected by the dead weight on journals which even with plain bearings is a small matter, while with ball or roller bearings, it is a very trifling consideration indeed.

35 I believe therefore that a pressure angle of $22\frac{1}{2}$ deg. can be adopted without fear of reduced efficiency in the transmission of power; that an addendum of $\frac{7}{8}$ module will give an ample arc of action for all combinations of gears between a 12-toothed pinion and a rack, and that true involute forms made to these constants will avoid the necessity for any empirical modifications and give results comparable with the best now obtained by such means.

TOOTH GEARING

By J. D. Steven,[1] Assoc. Mem. I. M. E.

ABSTRACT OF PAPER

A paper upon Tooth-Gearing was also presented at the same meeting by J. D. Steven, copies of which have been sent to the members of the Society. In this paper different methods of cutting gears were discussed and different systems of tooth design were compared, with the conclusion that conditions have so radically changed during the past two years that there is every reason to consider the question with an open mind. If a new form of tooth is desirable, he believed that a stub tooth with 20-deg. pressure angle would be a change in the right direction, because it can be used in its true form down to 12 teeth; it is a stronger form and is most commonly used; and a large proportion of its face does useful work.

[1] Messrs. E. G. Wrigley and Co., Soho, Birmingham, Eng.

DISCUSSION

Below is given the discussion presented at the meeting on the papers by Messrs. Lewis and Steven. This discussion will remain open until November 1 and additional contributions or comments are solicited from those interested.

P. V. VERNON[1] said he was present at the meeting as a patient and not as a doctor. He was connected with a firm of machine-tool makers, which had made many long and costly experiments with a view of determining, not what he might call academic points in the design of gearing, but for the purpose of obtaining a system of gears that would run silently at high speeds. His object in speaking was to ask that something might be done to discover the points that would give silence in gears, as until gears could be obtained which ran silently the question of making them interchangeable could hardly be profitably considered. The results of the experiments which his firm had made had been very small in the way of positive knowledge, but they had been very great negatively in the way of teaching them how very little anybody really knew about gearing. Among the negative results were the following: In the first place, it appeared that there was no general agreement among engineers or gear makers as to the best angle of pressure to use; there was no general agreement as to the proportions of gear teeth which were the best for quiet running; there was no reliable information on the strength of cut teeth made by modern methods in various materials and running at various speeds; there was no reliable information at present available on the effect of different materials on the strength, the noise, and the wear of gear teeth. It appeared, in fact, exceedingly doubtful whether it was possible to make an interchangeable system of gear teeth at all for quiet running.

2 Previous speakers had stated that the involute system was imperfect; engineers knew that the epicycloidal system was dead; and the results of the experiments which his own firm and others had made had proved that involute teeth did not give a quiet gear; that epicycloidal teeth did not give a quiet gear; that hobbed gears were not quiet, and that planed gears were not quiet. But there was one kind of gear which was quieter than all those, namely, the gear with "faked" teeth. It now was well known that the gears which gave the quietest results in running were those on which an empirical

[1] Mem.I.M.E., Messrs. Alfred Herbert, Coventry, Eng.

correction of the tooth form had been made, and so far as he knew the means by which that correction was made were kept secret, and were in the hands of one leading American firm, of which a representative was present at that meeting. When Ralph E. Flanders read his paper before the American society in December 1908, this same representative of that firm practically admitted that the only way to get quiet gears was to make an empirical correction on an experimental basis, and that the best thing for gear users to do was to trust to the experience of that firm, which he (Mr. Vernon) supposed meant to buy their cutters from them. He spoke at that meeting as a member of both the Institution of Mechanical Engineers and of The American Society of Mechanical Engineers, and wished to say that although the greatest credit was due to that firm for having made their investigations and experiments, it was not to the interest of gear users in America and in England that that knowledge should continue to remain in their hands alone. He hoped the remarks he had made would do good by bringing the committee of The American Society of Mechanical Engineers, who were considering gear questions, into touch with the committee of the Institution of Mechanical Engineers, so that investigations might be started with a view to ascertaining what were the corrections which needed to be made to ensure quiet tooth action, and so that such information, when obtained, could be placed in the hands of gear users in both countries.

C. R. Gabriel. This discussion was prepared at the request of the Meetings Committee of The American Society of Mechanical Engineers as a contribution upon standards for gears, and is submitted as a personal opinion, rather than a report, as the findings of the committee up to date have not established definite conclusions.

2 As stated, the committee received in reply to its circular letter, a large number of answers, but few contained comparative data and were largely expressions of theories or opinions, such as most engineers have advanced in explanation of the difficulties arising from improperly cut and noisy gears. The expression "improperly cut" is used advisedly, as from an experience covering a long period of years the writer believes that nine-tenths of all the trouble with noisy gears can be attributed to improper cutting.

3 As regards the adoption of a standard, the opinion of the writer is that the present so-called Brown and Sharpe standard will hold its own against any system for the general run of machinery; and that for many of the cases where special cutters or forms of teeth are used,

the benefits of such modifications are more imaginary than real. The answer to the difficulties arising where high speed, durability and efficiency are required, will be found in the substitution of better material and better methods of treating in the hardening process.

4 The present system is called by some an empirical one, but any carefully-worked out system will be so, as practice shows that decidedly better results are obtained by certain modifications of the theoretical tooth form. Teeth having an increase of pressure-angle and reduction of addenda to prevent undercutting on 12-tooth pinions would be subject to the same modifications which would put them in the empirical class, and any manufacturer of cutters would undoubtedly strive to improve his cutters regardless of whether they were strictly in accordance with theory.

5 The opinion is expressed that if such modifications are necessary, the methods should be known and clearly defined, so that makers of gears can use them and make their own cutters, if necessary. The advantages to the gear manufacturer of making his own cutters are not apparent to the writer, as the difficulties attending the making of form cutters would prove expensive in comparison with the convenience of using commercial stock cutters produced by reliable cutter manufacturers.

6 Attention has been called to the increasing number of gears cut by the hobbing process, and the difficulty of making such gears interchange with certain cut gears, thus demonstrating that the hobbing principle is the one to which all cutters should conform. This seems misleading, as the fact is not mentioned that this process has its limitations. It must not be overlooked that one great disadvantage arises from the difficulty experienced in producing accurate hobs.

7 The method of producing gear teeth by the shaping process has also been actively exploited, and the committee's attention has been called to the fact that it is an incidental feature of this process that the regular standard $14\frac{1}{2}$-deg. cutters (on account of the side strain involved) are less durable than the usual form of revolving cutter, especially under rapid production requirements, and in tough material. The proposed 20-deg. pressure angle stub tooth materially increases the strength of the reciprocating gear-shaper cutter. The writer also calls attention to the fact that the stub-tooth system would greatly reduce the difficulties in producing the hobs for the hobbing process referred to above, and believes the committee should be very conservative in its recommendations, particularly where they appear to favor some method or methods of producing gear teeth.

8 As regards the forms of teeth referred to in the paper as submitted by the other members of the committee, Mr. Fellows proposes to use an involute with 20-deg. pressure angle, and an addendum of $\frac{3}{4}$ of the modulus. Mr. Lewis proposes an involute with $22\frac{1}{2}$-deg. pressure angle, and an addendum of $\frac{7}{8}$ of the modulus. Both of these systems are subject to criticism for general adoption, (*a*) because of increased angular back-lash, as compared with the present $14\frac{1}{2}$-deg. standard for a like amount of wear; (*b*) decrease of addendum which is decidedly objectionable on the smaller pitches, as the liability of error in depth of tooth and center distance of shafts remains a constant for all pitches and sizes of gears, and the present height is none too great, being increased for these reasons for some purposes.

9 Further, it is claimed for the stub-tooth system that the arc of contact is equal to that of the $14\frac{1}{2}$-deg. Brown and Sharpe standard, and that consequently the excess of tooth height in the standard system produces unnecessary friction and wear, which is eliminated in the stub-tooth system. The fact is not mentioned that in the Brown and Sharpe system the height of tooth permits of being eased off towards the point, with the result that it causes no friction if the integrity of the tooth form is maintained, but as wear cannot be entirely eliminated this easing of the points of the teeth allows the engaging teeth to come into action without shock, thus reducing noise and the dynamic action which tends to destroy the gears by the contacting of the teeth and is similar to hammer blows. Also, in practice the standard length of tooth will conjugate itself with its mating gear, with the result that more than one set of teeth take the load. This may produce extra friction, as claimed, but with properly lubricated gears this is a negligible quantity, as the unit stresses are divided over more than one set of teeth. The stub tooth has greater strength, but, for the reason cited above, does not prove as satisfactory for all-round service.

10 In this connection, the writer wishes to refer to the situation in motor gears used in street-car service. From personal observation he has seen stub-tooth gears where this dynamic action seemed to have broken down the structure of the steel to such an extent that it came off in splinters where the points of the teeth seemed to have bitten the material away. The best results the writer has seen in this class of work are where gears and pinions have been cut to the Brown and Sharpe system and carbonized and case-hardened, with the result that after a year's use there was only glazing of the contact surfaces.

11 As to the matter of testing, the committee are united on its desirability, but the writer does not believe that anything definite or conclusive can be obtained from testing a few pairs of gears. The tests should cover a large number of gears of more than one pitch and of several different ratios of teeth. These tests should be so exhaustive as to leave no opportunity for engineers to feel that anything has been assumed. This means great expense and much time, and the writer would recommend that the coöperation of all interested parties be secured, that a fund be raised for the purpose of employing competent engineers to conduct exhaustive tests with suitable apparatus, and that all available materials be used, so that the differences of opinion as to the efficiency of the various systems in use may be set at rest as far as possible. Also this should establish reliable data for proportions of gears of the various materials, now mainly a matter of judgment with the designing engineer, from which more gear failures result than from any fault of the system of teeth used.

12 To sum up the situation, the writer believes that where the limits of an interchangeable system are from a 12-tooth pinion to a rack, the present system is superior to any which are primarily based on the generating principle, in which the pressure angle and tooth parts are proportioned to facilitate the methods of producing and strengthening the cutting tools, rather than to improve the interchangeability or more quiet running of the gears, as that is a fact still to be demonstrated. The circular letter sent out by the committee brought out the information from one concern that it had experimented exhaustively with the hobbing system for a long time, with various tooth forms, but had gone back to the Brown and Sharpe standard, for which they had now worked out a most satisfactory method of making the hobs. This is another example of the working out of an empirical system, and any satisfactory system covering the range stipulated, which, as before stated, must be an empirical one, had best be left to responsible cutter manufacturers whose methods combine the theoretical with the practical. The present Brown and Sharpe system has proved itself eminently satisfactory for many years, and without most thorough investigation should not be given a place second to any other. Also the introduction of an alternative standard or standards is to be deprecated on the ground of causing confusion and economic waste, due to the increased stock of cutters required for use on all types of machines.

13 In conclusion, the writer wishes to refer to the point raised in some of the replies to the committee's letter, that the question of

standardizing gearing is of the same nature as screw-thread standardization, and can be treated in the same way. Without referring to the conclusions reached by the Committee on Screw Threads, the deliberations of which consumed several years, the writer fails to see wherein the problems set forth by the two subjects can be compared. The screw-thread situation was one to be settled by common consent of all concerned, while the problem of gearing is one that is worthy the life work of any engineer, and involves much of the higher mathematics of engineering, as well as a thorough knowledge of applied kinematics. The fact that such a comparison has been made by some writers on the subject, only serves to show an utter lack of appreciation of the magnitude of the problem set before the committee for its consideration and recommendation to the Society.

LUTHER D. BURLINGAME said he appreciated all Mr. Vernon had said with regard to their policy at the Brown and Sharpe works, but he thought the point which had been raised was very well answered by the communication of Mr. Gabriel to which the members had just listened. Speaking for a manufacturer, at whose works for more than sixty years an extensive business in making gears and gear-cutters had been done, and which had had to deal with all phases of the gear question and meet the varying requirements of customers during those years, he could say they were not surprised that the different members of the committee of The American Society of Mechanical Engineers were not unanimous as to any plan which would bring about a change from the present standard. The thought seemed to be uppermost in the minds of those proposing a change that a standard should be adopted having a true involute for the entire length of the tooth, even if sacrifices in other directions must be made to accomplish that result. He believed the advantage, if any, of having the entire tooth a true involute was less than they would be led to believe by those advocating a change; indeed it was a matter yet to be demonstrated that the practical advantages of the form of tooth now in common use would not more than outweigh any theoretical advantages of a tooth wholly involute. The fact that the special requirements at the present time were for teeth varying both ways from the present standard would indicate that they now had a good average to meet ordinary working conditions. Their experience went to show that for quiet running for a possible variation in center distance and to avoid back-lash, a small pressure-angle and long teeth were of advantage, while for cases in which strength was the prime considera-

tion, even at the sacrifice of other advantages, the greater pressure angle and shorter teeth had their field of usefulness. Those varying needs were illustrated in automobile practice, where in some makes the cam-shaft gears were made with a small pressure angle and long teeth, while the driving gear was made with a larger pressure angle and shorter teeth, in each case for the reasons he had previously referred to.* That strength in the latter case was secured at some sacrifice was indicated by the illustration used by Mr. Gabriel in his communication, where he spoke of the injurious effects of the hammer-blow action when "stub-tooth" gears were used in street-car service. The same destructive action had been noticed in stub-tooth gears used for automobile service. An engineer who had for many years made a specialty of gearing, and who was an expert on that subject, and who had in recent years been with one of the leading automobile manufacturers of America, told him that he had repeatedly found the same destructive action spoken of by Mr. Gabriel when stub-tooth gears were used in various makes of automobiles.

2 Mr. Lewis, in referring to the preliminary tests made by the students of the Massachusetts Institute of Technology, said they showed that the loss by friction was varied to a greater extent by the length of the addendum than by the angle of obliquity.[1] As he understood, the tests to which Mr. Lewis referred were made entirely with roller bearings, thus reducing the bearing friction to a minimum. That friction, however, was one-third greater for the 20-deg. pressure angle than for the $14\frac{1}{2}$-deg. angle. With plain bearings, where the friction on the bearing would be much greater than with roller bearings, that factor favorable to the smaller pressure-angle might more than offset any advantage shown by the tests in favor of the greater pressure-angle as reducing friction. As plain bearings were more extensively used than roller bearings, the tests could not be considered complete without securing full data when using plain bearings. He agreed fully with Mr. Gabriel as to the need of exhaustive tests by competent engineers before any result could be accepted as authoritative.

3 He wished to take the present opportunity of endorsing what Mr. Gabriel had said relating to other phases of the subject. He did not believe that any modification of the present standard which would require more cutters in a set to ensure good running gears would meet with favour among manufacturers, as it would add directly to the cost of the equipment. All of the plans suggested in Mr.

[1] Given verbally by Mr. Lewis when presenting his paper.—EDITOR

Lewis's discussion were such as would require extra cutters in the set. Neither did he believe that any plan would be generally approved which would increase the amount of back-lash with a given variation in centre distance or with a given amount of wear. Unless some very important advantages could be shown for some new system yet to be suggested, it would seem unwise to bring about the confusion which would result from a change not fully endorsed by the manufacturing public.

DANIEL ADAMSON[1] wrote that, as no doubt Mr. Steven's paper was intended to assist the proposed joint committee of the English Institution and The American Society of Mechanical Engineers, he thought that one or two important processes for the production of gearing, which were not mentioned, should be referred to. There was no mention of the Sunderland method of generating spur gearing, the machines for which were at present being made by J. Parkinson and Sons, Canal Iron Works, Shipley. In this system the cutter took the form of a rack, and it was found that gears could be cut very much more rapidly than by rotary cutters, and the further advantage was that the cutters were much simpler and cheaper. Only one cutter was required for any number of teeth of a given pitch and the teeth were all correctly cut, as distinguished from the rotary-cutter system, wherein each cutter was only correct for the exact number of teeth for which it was designed.

2 He thought it seemed hardly fair (at the bottom of p. 4) to quote as a "difficulty" the fact that gears produced in hobbing machines by hobs cut to Brown and Sharpe standard would be unsuitable for running with those cut to other standard shapes. This was the fault of the difference in the so-called "standards" and was not the fault of the hobbing method, but was in fact the main argument for an early standardization of all spur-teeth upon a definite system.

3 Coming to worm-gears, there was no mention of the Gibson system, which was described in the Proceedings of the North-East Coast Institution of Engineers and Shipbuilders in 1896-1897, Vol. 13. This system should certainly have some consideration, as he believed more worm-gears had been cut by this method and by the subsequently improved method introduced by Hy. Wallwork and Co., of Manchester, the licensees of the Gibson system, and known as the Wallwork-McClean method, than by any other system.

[1] Messrs. Joseph Adamson and Co., Hyde, Eng.

4 In regard to the question of models of teeth (p. 6), there ought to be some consideration given to Lasche's paper in the Zeitschrift des Vereines deutscher Ingenieure for 1899, Vol. 43, wherein was described a system recommended by its author for the use of the A. E. G. This system was based upon a 15-deg. angle of pressure with an addendum of $1\frac{1}{2}$ module for the driver and $\frac{1}{2}$ module for the driven wheel, and bore some resemblance to the Bilgram system referred to in Wilfred Lewis's paper. There were many advantages claimed for it, namely, a stronger shape for the teeth of the pinion, a shorter arc of approach, and (when a rotary cutter was used) a more effective cutting shape, the radial flanks of the pinion teeth being very much shortened.

5 No paper on tooth gearing could be considered complete without at least a reference to Michael Longridge, Member of Council, who gave, in his annual reports for 1887-1888 and 1891, examples of very short teeth for large driving wheels. It was principally upon the basis of those reports that Mr. Adamson's firm adopted a shortened form of tooth in 1897, full particulars of which would be found in Vol. 24, issue No. 25 of the *American Machinist* of 1901, and which he was pleased to notice had the same length of addendum as the "stub-tooth" given in Table 1 of Mr. Steven's paper, namely, 0.25 of the circular pitch.

6 In conclusion, he sincerely trusted that the proposed joint committee would be encouraged by the discussion on these papers to push forward the important question of standardization of gear teeth, and to put it upon such a basis that satisfactory gear teeth could be produced in the future from a definite formula, without the use of any empirical correction.

THOMAS HUMPAGE[1] said that he thought much credit was due to Brown and Sharpe for standardizing their system of toothed gearing, which was known and used throughout the world. It was, however, introduced before the generating machine was in use, and it must give place to something better now. In view of the great advances that had been made during the past ten years in high-speed machinery owing to the introduction of high-speed steel and the great demand for silent running gears, the writer thought that we were undoubtedly behind the times in producing correctly-formed teeth to run quietly at these high speeds with great pressure, and he welcomed

[1] Mem. I.M.E., Messrs. Humpage, Thompson and Hardy, Jacob St., Bristol, Eng.

any improvements that might be adopted to attain this end. The Brown and Sharpe system should be left alone, as it would always be retained by certain manufacturers because it was good enough for their class of work.

2 The rack being the basis of all generated involute gears, the teeth should have straight sides, leaving the usual fillets at their roots below the working depth; and no deviation from this form should be made. A 12-tooth pinion should be the minimum for general purposes, because it has proved best in the past. For quietness and long life at high speeds, it was most essential that there should be no undercutting, and that the involute should extend to the whole working depth of the tooth. To generate these teeth on a 12-tooth pinion, the cutter or hob must have teeth with sides inclined at an angle of 22 deg., as a less angle would shorten the length of the involute and cause undercutting. As regards the addendum and dedendum, he quite agreed with Mr. Lewis's suggestions. These teeth could be generated on either of the following generating machines—an automatic hobbing machine, a Bilgram, Reinecker or Sunderland spur-gear planer, also on a Fellows gear shaper, providing the teeth of the cutter were ground to a true involute form down to the working depth of the teeth. They could also be cut on any machine of the Brown and Sharpe type with disc cutters made the exact shape of the space between the teeth. All wheels cut on these machines would then be interchangeable with each other. To facilitate repairs, it would also be a great advantage if manufacturers would stamp the pressure angle on the rims of the gears they cut, for example—$14\frac{1}{2}$ deg. Brown and Sharpe, or 22 deg.

3 It was, however, most unlikely that all the branches of the engineering trade using gears would accept any one standard pressure angle only. Taking the automobile industry, for instance, a 20-tooth minimum pinion was adopted by leading makers, and for change-speed gears a pressure angle of about 18 deg. would be sufficient to extend the involute to the working depth of the teeth without undercutting. It would also enable all existing machines in motor-car works to cut interchangeable gears, providing their cutters were formed on an 18 deg. angle basis.

4 An 18 deg. angle involute tooth was much stronger than the $14\frac{1}{2}$ deg. Brown and Sharpe standard tooth, which was the one generally adopted by the motor-car manufacturers, and owing to the involute being extended the wheels would run more quietly and last longer. The outward pressure would, however, be slightly increased above

the Brown and Sharpe standard, but it would be less than a 22-deg. angle. Again, in the printing machine and many other industries, the smallest pinions had more than twelve teeth, and a pressure angle of less than 22 deg. would be a great advantage owing to the fact that, when wear had taken place there was not so much back-lash and the machines registered more accurately.

5 The above suggestions appeared to the writer to be the first steps in the right direction leading up to the far more important and the much-desired object of producing perfectly silent running gears. In the writer's paper on Spur Gearing, read before the Institution at the Summer Meeting in Bristol of July 1908, he described a method of grinding the involute teeth of gear wheels. Since then he had continued his experiments, which he believed to be the best way of producing correct involute teeth and silent-running gears, and hoped shortly to make the results known.

R. M. NEILSON[1] wrote that, in view of the necessity of having maximum efficiency and a small angle of pressure in many cases where high powers were required to be transmitted, as, for example, in the propulsion of ships, it would appear to be advisable to have two standard types of teeth—one being of a nature to allow of the teeth being cut with a disc cutter for any number of teeth per wheel down to about 12 (or possibly less than 12), while the other would be designed for maximum efficiency and contact and a small angle of pressure, but would not be of a nature to allow of execution by a disc cutter except for wheels having a relatively large number of teeth. To attempt to make one standard serve in all cases would force the employment of non-standard teeth in many cases.

J. R. WILLIAMS[2] wrote that in his opinion the suggested increase in the angle of pressure, although good, both from the interchangeable point of view and increased strength of the teeth, would have a tendency to form shoulders on the working base of the teeth; this would, of course, increase the pressure on the bearings to a greater extent than the theoretical pressure, due purely to the increased angle, and would have to be allowed for when calculating the bearing surfaces for the shafts.

[1] Mem. I.M.E., 45 Hope St., Glasgow, Scotland.

[2] Asso. Mem. I.M.E., Tramway Power Station, Kelham Island, Sheffield, Eng.

A COMPARISON OF LATHE HEADSTOCK CHARACTERISTICS

By Prof. Walter Rautenstrauch, Published in The Journal for June 1910

ABSTRACT OF PAPER

The paper is concerned with the determination of the adaptability of a number of engine lathes to the economic performance of a standard task. The method of comparison, devised by Dr. J. T. Nicolson and Dempster Smith, of Manchester, England, consists in determining the extent to which each speed-torque combination possible in a given lathe lends itself to removing the greatest weight of shavings in a given time from a given diameter of work. The standard task assigned to each lathe is taking a predetermined area of cut on all diameters of work of mild and soft steel pieces, with high-heat steels. The results of the comparison are not intended to justify or condemn a lathe for general purpose work, but to exhibit the attempts of different manufacturers to meet the conditions favorable to the use of the high-heat steels. They show, among other things, that a more rational means of spacing speed-torque combinations is needed. Many of the lathes examined are shown to have a number of speed-torque combinations which may be dispensed with, as far as their adaptability to more economic weight removal is concerned, no matter what the standard of comparison may be.

DISCUSSION

Carl G. Barth. Professor Rautenstrauch's paper interests me as an evidence of the new attitude of the university and college professors towards the problems of the practical engineer, for it is my impression that until recently they concerned themselves only with the most general theories except for machinery connected with the generation of power.

2 I regret to say, however, that the method of analysis adopted in this paper is hopeless as a means to the end in view, for the same reason that Mr. Taylor's labors for twenty years previous to the perfection of slide-rules for the scientific running of machine tools never fully resulted in what he was after, namely, a ready and accurate means of predetermining just how efficiently metal might be removed by a lathe or kindred machine tool from any piece of work at all fit to be handled in such a machine. This paper practically reattempts a theoretical solution of the basic part of the problem

Mr. Taylor and his earlier associates tried to solve, and that the present attempt has failed, except in calling attention to the desirability of a more wide-spread realization of both the problem and its solution, is indeed only natural.

3 The problem is too complex for any ordinary mathematical analysis and requires for its solution a slide-rule of the kind just referred to, on which the great number of constants and variables entering into the problem may be so happily arranged with relation to each other that the solution of any problem involving a part or all of these is solved by a moment's inspection.

4 Besides the impossibility of utilizing the method of analysis proposed in this paper for practical purposes, it would appear that the author has started out with a misconception of how a lathe is properly used in practice. He assumes that the cut a lathe is capable of taking on its full swing diameter is all that it is ever called upon to take, while the fact is that when used for the mere purpose of removing metal, a lathe is properly confined to work on diameters not exceeding the swing over its carriage, or even less. The maximum cut a lathe can take is limited only by the strength of the cutting tool when this is properly held in its post with a minimum of overhang. A lathe is therefore most efficiently utilized for roughing when the work routed to it is of such medium diameters and lengths that the maximum cut the tool can carry does not, on the one hand, put an undue stress on the weakest driving gear, nor, on the other hand, tend to spring the work beyond the elastic limit of its material.

5 Before taking steps to decide how low to make the minimum spindle speed of a lathe, it is therefore necessary to determine the maximum pressure to which we will subject its cutting tool and the ratio of this to the permissible pressure on the teeth of the largest spindle gear. From this is obtained the largest diameter on which the maximum cut can be taken which the tool itself will safely bear. Then can be determined the size of this cut and the cutting speed at which it may be taken on the hardest material on which it is intended to be taken and still have the tool last an economical length of time before burning out or wearing down to an undue degree without burning out, as for extra hard materials. Rarely, however, can the speeding of a lathe be determined with regard to roughing work only, since this is usually simply one of many factors that have to be considered.

6 To determine the size of motor to provide for a lathe, the softest

material on which it is desired to take the heaviest cut the tool will stand must be decided and the cutting speed at which this may be taken with an economical tool life. The minimum revolutions of belt-driven as well as motor-driven lathes and the size motor to provide for the latter are most readily determined by means of special slide-rules for the purpose, on the basis of the information bearing on the subject contained in Mr. Taylor's paper. On the Art of Cutting Metals, but approximate formulae for the purpose are easily developed.

THE AUTHOR. It is to be regretted that the presentation of this paper has failed to call forth from the designers of lathes and similar machines a series of discussions which would throw additional light on the problem. It is hoped that this matter will not rest where it is, but that all having any knowledge of this subject will eventually contribute frankly and freely some item of value.

2 Mr. Barth's remarks are of much interest because of his long experience in this field. He does not confine himself to the body of the argument, however, although his contribution is of importance in that it presents a different point of view on what properly constitutes a basis for founding a method of comparison.

EXPERIMENTAL ANALYSIS OF A FRICTION CLUTCH COUPLING

By Prof. Wm. T. Magruder, Published in The Journal for January 1910

ABSTRACT OF PAPER

The paper gives the results of five lines of investigation on: (*a*) a dynamic clutch-testing machine; (*b*) a static apparatus, using 24-in. four-jaw friction clutch couplings from stock.

The solid clutch broke at 200 h.p. at 100 r.p.m., when the load was gradually applied, and at 110 h.p. at 100 r.p.m. for a pick-up load; the split clutch broke at 140 h.p. at 100 r.p.m. with the load gradually applied. A maximum axial thrust on the collar of 600 lb. may be required, or 115 lb. at the end of a one-to-five shifter lever, creating a total pressure of 8000 lb. by the shoes on the ring, or 50 lb. per sq. in.

DISCUSSION

H. J. Smith. The author gives three methods of making friction clutch tests:

a Static test to determine the force required to cause the clutch to slip with a given force on the shifter lever.

b The clutch in motion with the load gradually applied.

c The clutch in motion with the load suddenly applied, to determine the pick-up load capacity of the clutch.

2 While the first two methods are interesting, the results do not apply directly to actual working conditions. The pick-up test alone determines the capacity of the clutch and its factor of safety. Both experience and the results of numerous tests show that the pick-up load is extremely variable, depending on the class of work. In some instances in actual practice it is more than 210 per cent above the normal running load of the clutch. From this it is evident that both in the rating and in the selection of a clutch for a given service, the pick-up load is the controlling factor.

3 When a clutch under test has its load gradually applied, the transmitting capacity of the clutch is at its best, and its factor of safety highest. When called upon to pick up a dead load suddenly, the effect on the clutch is that of impact, and the factor of safety is reduced very often to a dangerous degree.

4 It will be seen from test I in Table 1 of the paper, the only recorded tests upon a split clutch, which is the weaker type of construction, that with a gradually-applied load the clutch transmitted 146 h.p. and broke. The stronger construction—the solid clutch—carried 187 h.p. and broke. Now note that this latter solid clutch carried only 102 h.p. on a suddenly applied, or "pick-up" load, and broke.

5 On the basis named by the author for gradually applied loads, that the four-arm split clutch will carry 78.5 per cent of the breaking load carried by the four-arm solid clutch, the split clutch would have broken on a pick-up load of about 80 h.p. This is in line with extensive prony brake tests made by me in 1908 and 1909, which showed for pick-up loads that 80 h.p. at 93 r.p.m. is about the rupture capacity of 24-in. split clutches and about 100 h.p. for 24-in. solid design clutches, as made by different manufacturers.

6 I must, however, take exception to the author's final conclusion that a factor of safety of 10 exists in clutch manufacture. This is far too high. In the first place the coefficient of rest is greater than the coefficient of motion, and we must not forget that the transmitting capacity of a clutch is dependent upon its ability to pick up its load from rest. In practice the dead load of the prony brake is replaced by the friction and inertia of large masses, such as occur in starting into motion long lines of shafting, flywheels, pumps, attrition mills, ball mills, tube mills, or other similar machinery.

7 During the time of slippage under the pick-up loads the coefficient of friction is lowest and the wear on the wood shoes greatest. Once in motion the friction of rest, which exists between the clamped members of the clutch, insures the continued transmission of any load that can be picked up.

8 In the second place the rating of a clutch of any size should be based on the split construction. In the test under consideration the rupture capacity of the 24-in. split clutch is about 90 h.p. at 100 r.p.m. The rating of the makers is 19 h.p. at the same speed. The factor of safety, therefore, is less than five.

9 In all tests made by me the final rupture occurred at the keyway of the clutch hub; the bolting lugs and bolts gave way at the same moment and the ring was broken only by the thrust of the broken clutch mechanism.

10 There is no doubt that the transmitting capacity of a clutch increases practically proportionately with the number of arms; on the other hand it is equally certain, from a study of design, that the factor of safety will vary with the number of arms. My own opinion

of the factor of safety of a Hill clutch of four-arm construction is that each arm, independent of the others, should be able to carry safely the rated capacity of the clutch at 100 r.p.m. This gives a factor of safety of four. For a three-arm clutch in the same manner the factor of safety is three, but for a six-arm clutch it rarely exceeds four. This is because a six-arm clutch has a no stronger hub construction than a four-arm clutch on account of the congested design.

11 In a series of tests by Prof. R. G. Dukes of the Case School of Applied Science, Cleveland, results were obtained upon the ultimate capacities of different types and makes of clutches. In the following table is given the performance of a 24-in. split positive-action Hill clutch, which, however, was not pushed to its greatest capacity on account of lack of power on the testing stand:

TABLE 1 TESTS ON 24-IN. POSITIVE ACTION HILL CLUTCH CATALOGUE RATING 48 H.P. AT 100 R.P.M.

Number of Test	Thrust on Collar, lb.	Maximum Brake Load, lb.	Horsepower at 100 r.p.m.	Remarks
1	258	540	74	Clutch slipped
2	334	830	113	Clutch slipped
3	410	880	120	Clutch slipped
4	544	1030	141	Limit of Testing Stand

12 To illustrate the enormous power-transmitting capacity of the ring type of clutch, the following test is of interest. It was made on a 24-in. No. 60 Smith type Hill split clutch of very heavy construction. It is rated at 60 h.p. at 100 r.p.m. Fig. 1 shows the design of the clutch, which is positive in its action, no springs being required for disengaging the jaws.

13 The test was conducted as follows: The power transmitted was measured by a prony brake and platform scales. The speed of the clutch shaft was maintained at 93 r.p.m. A spring scale measured the force applied to the hand lever. The wood shoes were first worn to a fit on the ring. Then the jaws were adjusted so that a certain pressure on the hand lever engaged the clutch. Next the brake band was tightened and the clutch engaged, forcibly picking up the dead load from rest. Repeated trials gave the maximum brake load which the clutch would pick up without slipping. When a slip occurred the jaw adjustment was increased to give larger hand lever scale readings, and corresponding higher brake readings. The final read-

ing of the transmitting capacity was limited only by the available power. The results are given in Table 2.

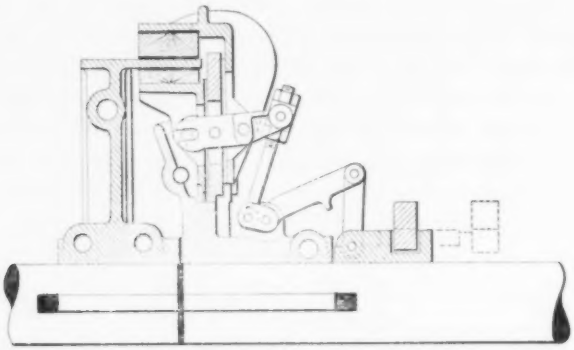


FIG. 1 24-IN. NO. 60 HILL SPLIT CLUTCH (SMITH TYPE)

14 It will be noted that these results were obtained with a split clutch starting from absolute rest—the true pick-up load—and bringing up to speed at the stated loads. No breakage occurred. It is the result of correct toggle action and rigidity of construction.

TABLE 2 TESTS ON 24-IN. NO. 60 HILL SPLIT CLUTCH (SMITH TYPE)

CATALOGUE RATING 60 H.P. AT 100 R.P.M.

NUMBER OF TEST	THRUST ON COLLAR, LB.	NET MAXIMUM BRAKE LOAD, LB.	HORSEPOWER AT 100 R.P.M.	REMARKS
1	188	365	58	
2	258	527	84	Clutch slipped on
3	312	728	116	further tightening
4	380	878	140	brake
5	456	1087	173	
6	740	1430	228	Limit of testing stand

Prony Brake Arm 100-in. long.

15 It is of interest to note also that the ring type clutch described in the paper was the invention of Harry W. Hill (February 10, 1885). In its fundamental construction two members are required: a toggle-locked friction clamp and a friction ring. To preserve the balance of the clutch and increase its capacity, additional friction clamps are added, making two, three, four or six arm clutches.

G. N. VanDerhoef. The real point in connection with friction clutches is undoubtedly the picking-up of the load, and these tests seem to lack in that they do not give the mass that is started from rest and the time required to accelerate it to uniform velocity. That is the duty the clutch has to perform and is the real measure of the strength of the clutch. The time should be given in seconds, or in fractions of seconds. When this is done we will have some idea of the relative value of friction clutches.

H. J. Smith. In regard to the time element in tests of friction clutches, I know that in Professor Duke's report and in the tests I conducted myself, it was found best to throw in the clutch just as rapidly as possible. In any case, on a pick-up load, clutches of all types were thrown in as quickly as it could be done by the impulse of two men on the hand lever.

E. P. Haines. In connection with the time taken to start a load, take the case of a gas engine in a machine shop. If the clutch is thrown in quickly, the gas engine will stop, and if it is thrown in too slowly, the clutch will burn out unless it has ample surface to prevent overheating. The time taken to apply the load should be between the two extremes.

H. J. Smith. Through this system of testing I have evolved the following rules: for single-cylinder gas engines, use a clutch of double the capacity at a given speed; for double and triple-cylinder gas engines, use one rated at from 25 to 50 per cent above the capacity desired; with engines of four or more cylinders, use a clutch of the rated capacity for steam engine practice.

Oberlin Smith said that clutches should always be thrown in as quickly as possible, especially if they have wooden or fiber shoes, except where gas engines are used for power. He has found that clutches as a rule are built too small in diameter, necessitating severe clamping to obtain sufficient friction to carry the load. This is sometimes necessary where the diameter must be restricted, but in general large diameter should be employed to obtain speed and to reduce the pressure on the clamps. He stated that his experience had been mostly with clutches used for starting presses, where the inertia of large shafts and heavy chains of gearing impose severe strains on them. Under such conditions the clutch surface is run at the highest

safe speed, about a mile a minute, in order to obtain the greatest benefit from the friction.

2 Many of the friction clutches in the market are excellent but complicated. They require a great deal of mechanism to give enormous pressure at the friction surface. For driving heavy presses a much simpler form of clutch is preferable and the old cone form has been adopted. By making the surface wide and large in circumference a very simple toggle mechanism is sufficient.

THE AUTHOR. In reply to H. J. Smith's remarks, I would say that every one is aware that a split pulley is not as strong as a solid one, and for that reason the number of split friction clutch-couplings sold is probably less than ten per cent of the number of solid ones made and sold.

2 In reply to Mr. VanDerhoef, it should be said that the mass that was started from rest is shown in Fig. 2. The 48-in. by 24-in. brake pulley weighed about 2800 lb. No automatic and autographic apparatus was available for recording the time required suddenly to apply the load. The instructions were to throw the load on as quickly as possible, and this was done. To describe this action as accurately as possible, it is called in Table 5 a suddenly applied load rather than a pick-up load. Just what fraction of a second this was each one can judge from his own experience. The term pick-up load is quite indefinite as to the time element and simply describes the starting into motion of a shaft, etc., which was at rest and with an indefinite amount of slipping of the friction members.

3 Replying to Oberlin Smith, I would say the intensity of pressure of the shoes on the ring was about 50 lb. per sq. in. with this clutch (see Par. 37 *Bd*).

TEST OF A 15,000-KW. STEAM-ENGINE-TURBINE UNIT

By H. G. Stott and R. J. S. Pigott, New York

This paper was published in The Journal for March 1910 and relates to tests of low-pressure steam turbines installed at the 59th Street Station of the Interborough Rapid Transit Company of New York, originally equipped with engines of the Manhattan type to which vertical low-pressure turbines and independent generators were added. The discussion upon the paper presented at the meeting and contributed later was published in The Journal for August. The following addition to the discussion is given by the author in reply to an inquiry as to the advisability of installing a new plant with a combination unit of engine and turbine.

The Author. In reply to the question by Max Rotter as to whether it would be advisable to install in a new plant such a combination as described in the paper, the accompanying diagram (Fig. 1), showing the factors entering into the cost of power, will answer his question. For example, with a low load factor, it is quite evident that the all-important point is to keep down the fixed charges for when the load factor is less than 20 per cent, the fixed charges are of vastly greater importance than any possible gain of efficiency due to a better type of prime mover. This is true of any plant, and the curve shows the futility of attempting to carry peak loads by means of gas engines, water power or any other prime mover necessitating a large investment per kilowatt. On the other hand, with loads having a load factor of over 60 per cent especially where fuel cost is high, the investment factor is of relatively small importance, and the all-important matter is to keep down the maintenance and operating charges by using the most efficient type of plant obtainable, provided that it is at the same time thoroughly reliable.

2 The total cost of power can be obtained from these curves by simply taking the sum of the ordinates above and below the axis for any load factor desired..

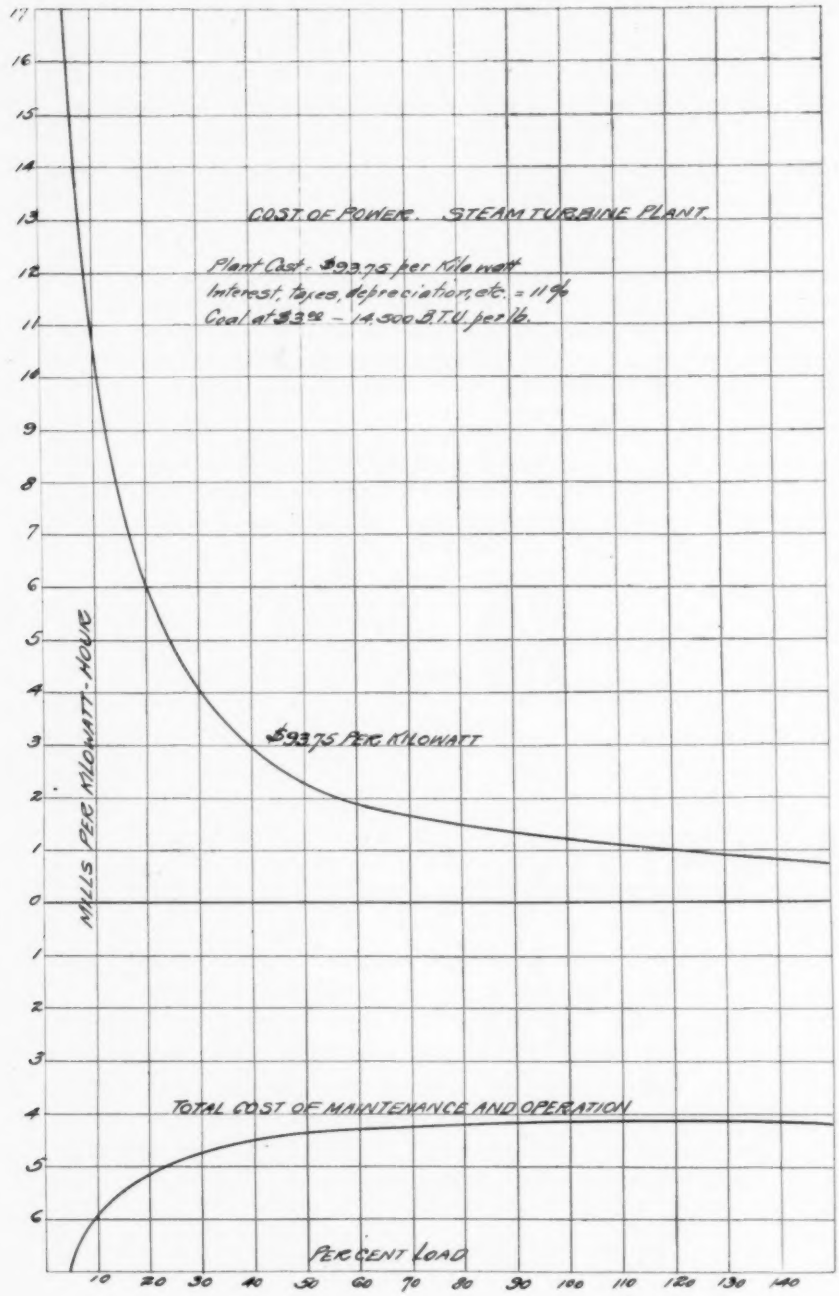


FIG. 1 TEST OF A 15,000-KW. STEAM-ENGINE-TURBINE UNIT
Diagram of Factors entering into the Cost of Power

GENERAL NOTES

AMERICAN SOCIETY OF CIVIL ENGINEERS

At the monthly meeting of the American Society of Civil Engineers, held on September 21, 1910, papers were presented on Long Island Approaches to the East River Tunnel, on The Sunnyside Yard, and on The Lining on the Four Permanent Shafts of the East River Tunnel.

AMERICAN SOCIETY OF ENGINEERING CONTRACTORS

The annual convention of the American Society of Engineering Contractors was held in St. Louis, Mo., September 27–29, 1910, in the Coliseum. Papers were delivered by J. B. Goldsborough and Edward Wegmann, both of New York, on Dam Construction for City Water Supplies; and by George C. Warren, of Boston, on Work Preliminary to Street Paving and Road Work. A banquet was held during the convention and several sightseeing trips made to important engineering works in and around St. Louis.

NEW ENGLAND WATER WORKS ASSOCIATION

The twenty-ninth annual convention of the New England Water Works Association was held at Rochester, N. Y., September 21–23. Among the papers presented were: Pertinent Matters Relating to Water Works, Frederick Eltwood; Purification Plant of Rochester and Lake Ontario Water Company, Rochester, N. Y., James M. Caird; Depreciation in Water Works and Methods of Accounting therefor, Leonard Metcalf, Mem.Am.Soc.M.E.; The Use of the Water Ejector for Transporting Sand, Morris Knowles, Mem.Am.Soc. M.E., and John M. Rice.

NEW YORK RAILROAD CLUB

The first regular meeting of the New York Railroad Club for the fall season was held Friday evening, September 16, in the Engineering Societies Building, 29 West 39th Street. B. S. Hinckley, Engineer of Tests of the New York Central and Hudson River Railroad, presented a paper entitled the Testing Department of a Railroad.

NATIONAL ASSOCIATION OF GERMAN-AMERICAN TECHNOLOGISTS

The twenty-sixth annual convention of the National Association of German-American Technologists was held at Newark, N. J., September 1–5. Dr. L. H. Friedburg presented a paper on The Activity of Radium and Permanency of

the Element with Side-Lights on Transmutation and Alchemy. Excursions were made to the Edison Laboratory, West Orange, N. J., and to the Henry R. Worthington Hydraulic Works, Harrison, N. J.

ASSOCIATION OF RAILWAY ELECTRICAL ENGINEERS

At the convention of this association, held September 27–30 at the La Salle Hotel, Chicago, Ill., papers were presented on the following subjects: Electric Lighting of Railway Cars, Axle Generating Systems, Electric Traction on Trunk Lines, Motor Driven Machine Tools, and The Illumination of Railway Passenger Stations and Machine Shops.

ROADMASTERS AND MAINTENANCE OF WAY ASSOCIATION

The twenty-eighth annual convention of the Roadmasters and Maintenance of Way Association was held at the Great Northern Hotel, Chicago, Ill., September 13–16, with an attendance surpassing all former records. The following subjects were discussed: Proper Care of Track Material and Tools; Cattle Guards; New and Improved Appliances, including Ties; Standard Switch Target; Rail Fastenings, including Insulated Joints; Tie Plates and Anchors. A paper on Treated Timber was presented by J. L. Single.

THE COLORADO ELECTRIC LIGHT, POWER AND RAILWAY ASSOCIATION

The eighth annual convention of the Colorado Electric Light, Power and Railway Association was held at the Hotel Colorado, Glenwood Springs, Colo., September 21–23, and was the largest and most interesting meeting from a professional point of view that the association has ever held. Papers were read on the following subjects: Storage Batteries in Street Railway and Central Station Work, F. A. Warfield; Some Observations on Cultivating Friendly Relations with the Public, J. M. Connelly; Mazda Street Series Lighting, R. L. Jones; High and Low Tension Lines, J. C. Lawler; The Place of Rotary Condensers and Induction Generators in Transmission Systems, H. L. Aller. Reports were also received from committees appointed to investigate the Subject on Grounded Secondaries, Meters, Insurance, etc.

PERSONALS

E. T. Adams has resigned as chief engineer and manager of the gas and mill engine department of the Allis-Chalmers Co., West Allis, Wis., a position which he has held for the past five years.

H. P. Ahrnke has become associated with the Barrett Manufacturing Co., New York. He was formerly identified with the Pa. Tunnel & Terminal R. R. Co., New York.

Jos. E. Aue has become identified with the Snow Steam Pump Works, Buffalo, N. Y. He was formerly connected with the Pacific Gas and Electric Co., San Francisco, Cal.

Earle J. Banta, chief engineer of the Ohio Steam Shovel & Dredge Co., Cincinnati, O., has resigned his position to accept a similar position with the Vulcan Steam Shovel Co., Toledo, O.

Harry Z. Bixler has entered the employ of the Youngstown Steel Co., Youngstown, O., as chief engineer. Until recently he was associated with Worth Brothers Co., Coatesville, Pa.

Joseph B. Brady has resigned as assistant experimental engineer of the Hess-Bright Manufacturing Co., Philadelphia, Pa., to take up studies in electrical engineering at Purdue University, Lafayette, Ind.

Joseph Breslove recently associated with the engineering department of the Westinghouse Machine Co., Pittsburg, Pa., has been appointed mechanical engineer of the Nivison-Weiskopf Co., Cincinnati, O.

John F. Buckley, formerly general superintendent of the St. Marys Machine Co., St. Marys, O., has become general manager of the Buckley Specialty Co., Lima, O.

Carl F. Dietz of the firm of Dietz & Keely, Boston, Mass., has returned from Europe.

John A. Doane, formerly superintendent of shops of the Taylor Iron & Steel Co., High Bridge, N. J., is now associated with the American Optical Co., Southbridge, Mass., in a similar capacity.

Henry L. Doherty has resigned as president of the American Gas & Electrical Co He remains a director.

A. Hartill-Law, has been elected Mayor of Cairus, North Queensland.

J. E. Johnson, Jr., until recently general superintendent of the Thomas Division of the Republic Iron & Steel Co., Thomas, Ala., has accepted a position with the Lake Superior Iron & Chemical Co., Ashland, Wis., as manager.

Edward J. Kunze, formerly instructor in steam and gas engineering, University of Wisconsin, Madison, Wis., has been appointed assistant professor of mechanical engineering at the Michigan Agricultural College, East Lansing, Mich.

A. Lebrecht has resigned his position as chief engineer of the gas and oil engine department of the De La Vergne Machine Co., New York, N. Y., and has opened an office as consulting engineer in Nuremberg, Germany.

Ralph D. Mershon was elected president of the Alumni Association of the Ohio State University for the ensuing year.

Arthur F. Pitkin has accepted a position with the General Electric Co., Schenectady, N. Y., as assistant to the mechanical superintendent. He was formerly standard practice engineer of the automobile department of the American Locomotive Co., Providence, R. I.

H. F. J. Porter has been engaged during the past summer in an investigation of hygienic conditions and their effect on the efficiency of the employees in manufacturing establishments in Pittsburg for the Russell Sage Foundation.

L. R. Shallenberger, until recently superintendent of construction for the Griffin Wheel Co., with headquarters at Chicago, Ill., has opened an office in Paonia, Colo., for general engineering work.

P. S. Steenstrup, formerly general sales manager of the Hyatt Roller Bearing Co., Harrison, N. J., also secretary and director of the Motor and Accessory Manufacturers, has incorporated the Wayne Pacific Distributing Co., with general office in Medford, Ore., and with warehouses in Portland, Ore., and San Francisco, Cal.

H. Struckmann, until recently chief engineer and general manager of the Union Sand & Material Co., St. Louis, Mo., has become associated with the Iola Portland Cement Co., Iola, Kansas, in a similar capacity.

James W. Thomas, formerly of the engineering staff of Westinghouse, Church, Kerr & Co., New York, has been appointed mechanical engineer of the New England Engineering Co., New York.

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society, included in the Engineering Library. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary, Am. Soc. M. E.

L'AEROPHILE. Years 16 and 17, Nos. 1-21, 23-24. *Paris, 1908-09.* Gift of A. W. H. Griepe.

BOILER TROUBLES AND THEIR TREATMENT. By George W. Lord. *Philadelphia, 1910.*[1]

CATALOGUE OFFICIEL DES COLLECTIONS DU CONSERVATOIRE NATIONAL DES ARTS ET MÉTIERS. Pts. 4 and 6, 1908, 1910. *Paris, 1908, 1910.* Gift of Conservatoire National des Arts et Metiérs.

COLORADO SCIENTIFIC SOCIETY. Yearbook, 1882-1910. *Denver, 1910.* Gift of the society.

CONGRÈS (PREMIER) INTERNATIONAL DU FROID, PARIS, 1908. Comptes Rendus. Vols. 1-3. *Paris, 1908.*

THE ENGINEER. Published by the Engineering Students of the Pennsylvania State College. Vol. 3. No. 3. April 1910.

ILLINOIS SOCIETY OF ENGINEERS AND SURVEYORS. Twenty-fifth annual report. 1910. *Chicago, 1910.* Gift of the society.

EIN JAHR VERBAND DEUTSCHER DIPLOM-INGENIEURE. By Alexander Lange. Sonderabdruck Zeit. des Verbandes Deutscher Diplom-Ingenieure, pt. 16, 1910. *Berlin.* Gift of the author.

MCGILL COLLEGE AND UNIVERSITY. Annual Calendar, 1910-1911. *Montreal, 1910.* Gift of the university.

METALLOGRAPHIE. By W. Guertler. Vol. 1. No. 4. *Berlin, Borntraeger, 1910.*

MOTOR UNION JOURNAL. Vol. 2. No. 6. July 1910. *London, 1910.*

OHIO ELECTRIC LIGHT ASSOCIATION. Programme, 16th annual convention, July 26-28, 1910, Cedar Point, Ohio.

ORGAN FÜR DIE FORTSCHRITTE DES EISENBAHNWESENS. General Index. Vol. 1-44. 1864-1907. *Wiesbaden, 1873-1908.*

PENNSYLVANIA, COLONIAL AND FEDERAL. A History. 1608-1900. Edited by H. M. Jenkins. *Philadelphia, 1902.*

RESISTANCE DE L'AIR. By G. Eiffel. *Paris, 1910.* Gift of the author.

SCHWEIZERISCHER INGENIEUR UND ARCHITEKTENVEREIN MITGLIEDERVERZEICHNIS. 1909. *Zurich, 1909.* Gift of the society.

SOCIETY FOR THE PROMOTION OF ENGINEERING EDUCATION. List of the Officers and Members, April, 1910.

SOCIETY OF ENGINEERS OF EASTERN NEW YORK. List of Members, 1910. *Albany, 1910.* Gift of the society.

[1] **Corrected Title. Published incorrectly in The Journal for July.**

TECHNIK UND WIRTSCHAFT. Yr. 1, pts. 1-5, 7-12. *Berlin, 1908.* Gift of A. W. H. Griepe.

TECHNIKER IN DER MODERNEN STAATSWIRTSCHAFT. By Alexander Lang. Sonderabdruck aus dem Zeit. des Verbandes Deutscher Diplom-Ingenieure, Pt. 1, 1910. *Berlin.* Gift of the author.

TRAVELING ENGINEERS ASSOCIATION. Report of 18th annual meeting Niagara Falls, Can., Aug. 16-18, 1910. Gift of the association.

JAMES WATT'S CONTRIBUTION TO THE ADVANCEMENT OF ENGINEERING. By Robert Caird. From Papers of the Greenock Philosophical Society. *Greenock, 1910.* Gift of the author.

WISCONSIN GAS ASSOCIATION. Proceedings. 1910. *Milwaukee, 1910.*

EXCHANGES

AMERICAN SOCIETY OF CIVIL ENGINEERS. Transactions. Vol. 68. *New York, 1910.*

INSTITUTION OF CIVIL ENGINEERS. Minutes of Proceedings. Vol. 180. *London, 1910.*

JIGS AND FIXTURES. By Einar Morin. Pts. 1-3. *New York, 1909.*

MEMORIAL DU GÉNIE MARITIME. Ministère de la Marine. Dec. 1908, May and June 1910. *Paris, 1910.*

NATIONAL ASSOCIATION OF COTTON MANUFACTURERS. Transactions. No. 85. *Boston, 1909.*

TRADE CATALOGUES

ALLIS-CHALMERS CO., *Milwaukee, Wis.* Bull. No. 1054. Steam Turbines and Generators, 43 pp.

H. G. BARR, *Worcester, Mass.* Multiple spindle sensitive drilling machines, 2 pp; Drilling and tapping machines, 1 p.

BROWN HOISTING MACHINERY CO., *Cleveland, O* Modern ore and coal handling machinery, 80 pp.

CLEVELAND AUTOMATIC MACHINE CO., *Cleveland, O.* Automatic machines, attachments and tools, 114 pp.

CONSOLIDATED SAFETY VALVE CO., *New York.* Safety valve capacity with tables of capacities of safety valves, by P. G. Darling, 31 pp.

HESS-BRIGHT MFG. CO., *Phila, Pa.* Adapter bearings for radial loads on line shafts, 1 p; Marine propeller ball bearing thrust block, 1 p.

IDEAL ELECTRIC & MFG. CO., *Mansfield, O.* Bull. No. 1051. Revolving field alternators, 11 pp.

JOHNS-MANVILLE CO., *New York.* Roofing Salesman, Aug. 1910, 8 pp.

LINK-BELT CO., *Phila, Pa.* Peck carrier for conveying of coal and ashes in power houses, 96 pp.

NATIONAL ELECTRIC LAMP ASSOC., *Cleveland, O.* Bull. 9 A. Cost of Light, by S. E. Doane, 19 pp; Bull. 12. Electric Lighting of Automobiles, 23 pp.

NILES-BEMENT-POND CO., *New York.* Progress Reporter, June 1910, 20 pp.

JOSEPH T. RYERSON CO., *New York.* Stock list of iron and steel supplies, 144 pp.

UNDERFEED STOKER CO. OF AMERICA, *Chicago, Ill.* Publicity Magazine, July and August, 30 pp.

VEEDER MFG. CO., *Hartford, Conn.* Odometers and tachodometers for automobiles, 15 pp; Counters for machines, cash registers, and other automatic machines, 31 pp; Castings, 14 pp; Cyclometers for bicycles and motorcycles, 16 pp.

WATSON-STILLMAN CO., *New York.* Twin volute turbine pumps, 76 pp.

WHEELER CONDENSER & ENGRG. CO., *Carteret, N. J.* Bull. 106. High vacuum surface condensers for steam turbines, 19 pp.

R. D. WOOD & CO., *Phila, Pa.* Theissen patent centrifugal contra-flow gas washer, 23 pp.

UNITED ENGINEERING SOCIETY

ENGINEERING RECORD. Vol. 24. *New York, 1891.*

KANSAS GAS, WATER, ELECTRIC LIGHT & STREET RAILWAY ASSOCIATION. 12th annual meeting, Wichita, Kan., Sept. 23-24, 1909. Gift of the association.

MICHIGAN GAS ASSOCIATION. Proceedings of the 18th annual meeting, 1909. Gift of the association.

MUNICIPAL ENGINEERING. Vols. 22-23. *New York, 1902.*

SANITARY ENGINEER. Vols. 12, 16. *New York, 1885, 1887.*

STATEMENT OF FACTS CONCERNING THE SO-CALLED BARCROFT APPRAISAL OF THE DETROIT UNITED RAILWAY LINES IN THE CITY OF DETROIT, AUGUST 1910. Gift of R. B. Rifenberick.

STATESMAN'S YEARBOOK, 1910. *London, Macmillan & Co., 1910.*

SURVEYORS INSTITUTION. List of Members, 1910. *Westminster, 1910.* Gift of the institution.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 12th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

047 Wanted by foundry and machine company near New York several capable estimators and draftsmen. Opportunities to advance in engineering department or to other departments.

048 High grade designing draftsman with large experience in steam engineering practice, for a permanent position with large manufacturing concern in Massachusetts. State age, education, experience and salary expected.

049 Instructor, machine drawing and design. Initial salary $1000 to $1200 for seven months, promotion assured to right man. Candidate should have good experience in shop and drafting room. Department M. E., School of Mining, Kingston, Ont., Canada.

050 A young man to assist in engineering department, $850 for the college year to start. Should have some drafting room experience, besides a college degree; work involves gas engine design. Would prefer to have man who has had some experience in this line. University of Minnesota.

MEN AVAILABLE

116 Mechanical engineer, practical technical graduate, at present employed as factory superintendent in an eastern manufacturing concern, would consider an opening in a similar capacity or in general engineering work.

117 Member, technical graduate, charge of large shops as superintendent and manager, also estimating cost systems, correspondence, etc., thorough experience in design, construction and economical operation of power and manufacturing plants. Several years experience as sales manager Corliss engines; also as general sales manager Water Tube Boilers. Desires position with chance to secure an interest. Engaged in special work at present but desires permanency.

118 Member, age 35, practical mechanic, technical education Cornell University, desires position as general superintendent or works manager, good hustler and organizer. Experience with the heaviest class of machinery.

119 Mechanical engineer, graduate of M. I. T., five years practical experience, three years chief engineer manufacturing concern, desires a responsible position with good opportunities for advancement. At present employed.

120 Mechanical engineer, fourteen years experience, heating, ventilating, power plant, gas engine and gas producer, machine tools; general engineering experience covers shop, drafting room, selling and experimental station, factory methods and system. Inventive and executive ability; speaks English, German and French. Salary, $2000. New York City or vicinity.

121 Junior, 30, seven years as draftsman and designer of gas engines, large units, mining and special machinery. One year systematizer, six months assistant master mechanic over maintenance gang of one hundred; six months secretary of refrigerating air-machine company. Has executive ability, successful with help, adaptable, ingenious, resourceful.

122 Young engineer, Stevens graduate; specialized in power plant equipment, condensers and cooling towers, desires position leading to administrative responsibility.

123 Young engineer, Stevens graduate. Recently assistant to consulting civil engineer. Would like position with manufacturing concern. Junior Member.

124 Member, 30 years experience, designing, constructing, selling and installing general machinery for factories, railroads and sugar plantations, in Cuba, Mexico and South America, desires position of responsibility with company requiring service of experienced mechanical engineer. Speaks Spanish and is familiar with export business.

125 Junior Member, technical graduate, eight years experience in engineering and manufacturing, at present chief draftsman in large manufacturing concern in Michigan, desires position in the East.

126 Member having extensive executive and mechanical experience desires to secure a position in Greater New York as sales or purchasing engineer; large experience in designing, selling and buying sugar machinery.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ABRAHAMS, M. Landa (Junior, 1910), U. S. Engr.'s Office, Federal Bldg., Chicago, Ill.

ADAMS, Edward T. (1899), Milwaukee Club, Milwaukee, Wis.

ADAMS, Kilburn E. (Junior, 1908), Mech. Engr., Boston & Albany R. R., Rm. 372, South Sta., Boston, and *for mail*, 629 Watertown St., Newtonville, Mass.

AHRNKE, H. P. (Junior, 1902), Barrett Mfg. Co., 17 Battery Pl., New York, N. Y.

AUE, Joseph E. (1899), Snow Steam Pump Wks., Buffalo, N. Y.

AUTENRIETH, George C. (Junior, 1908), Instr., Descriptive Geometry and Drawing, College of the City of N. Y., New York, N. Y., and *for mail*, 5 Duer Pl., Highwood Park, Weehawken, N. J.

BANTA, Earle Jackson (1907), Ch. Engr., Vulcan Steam Shovel Co., Toledo, O.

BENNET, Orville G., Jr. (Junior, 1907), Okonite Co., 253 Broadway, New York, N. Y.

BLAKE, Clinton F. (1901), R. F. D. 6, Box 112, Oregon City, Clackamass Co., Ore.

BRADY, Joseph Benjamin (Junior, 1910), 218 Fowler Ave., West Lafayette, Ind.

BRAMAN, Samuel N. (Junior, 1902), 10 Newbury St., Boston, Mass.

BUCKLEY, John Francis (1906), Genl. Mgr., Buckley Specialty Co., Lima, O

DAMON, Walter Henry (1909), Supt. of Generating, United Elec. Light Co., 32 Bancroft St., Springfield, Mass.

DAY, Charles (Junior, 1902), Member of Firm, Dodge, Day & Zimmermann, Engrs., Wash. Bldg., 608 Chestnut St., Philadelphia, and 5021 Schuyler St., Germantown, Philadelphia, Pa.

DOANE, John Appleton (1910), Supt. of Mch. Shops, Am. Optical Co., Southbridge, Mass.

DODGE, Kern (Junior, 1902), Member of Firm, Dodge, Day & Zimmermann, Engrs., Wash. Bldg., 608 Chestnut St., Philadelphia, and 5135 Pulaski Ave., Germantown, Philadelphia, Pa.

DREYFUS, Theo. Frank (Junior, 1903), 895 W. Bancroft St., Toledo, O.

EVANS, Henry O. (1896), Contr. and Designing Engr. of Special Mchy., 566 Drexel Bldg., Philadelphia, Pa.

FLAGG, Stanley G., Jr. (1891), S. G. Flagg & Co., Morris Bldg., 1421 Chestnut St., Philadelphia, Pa.

FLETEMEYER, Louis H. (Junior, 1906), Ch. Draftsman, Canada Fdy. Co., Ltd., and *for mail*, 519 Brock Ave., Toronto, Ont., Canada.

FROST, Harwood (1910), 1636 Monadnock Blk., Chicago, Ill.

GREUL, W. Herman (1907), Otis Elev. Co., 17 Battery Pl., New York, N. Y.

HAGAR, Arthur Percy (Associate, 1907), Engr., Marine Dept., Safety Car Htg. & Ltg. Co., 206 Erie St., Jersey City, and *for mail*, 716 Cliften Ave., Newark, N. J.

HANSEN, T. H. C. (1905), Wire Specialty Wks., 549 Fulton St., and *for mail*, 2429 Leland Ave., Chicago, Ill.

HAZELTON, William S. (1909), 247 Hancock Ave., E., Detroit, Mich.

HENDERSON, Geo. R. (1890), Cons. Mech. Engr., 500 N. Broad St., Philadelphia, Pa.

HENES, Harry W. (Junior, 1909), A. Bolter's Sons, Structural Steel, Rm. 612, 84 La Salle St., Chicago, Ill.

HIBBARD, H. Wade (1891; 1894), Prof. Mech. Engrg., Univ. of Mo., 1307 Kaiser Ave., Columbia, Mo.

HILL, Alfred J. (Associate, 1902), 1330 Castillo St., Santa Barbara, Cal.

JOHNSON, J. E., Jr. (1896; 1901), Mgr., Lake Superior Iron & Chem. Co., Ashland, Wis.

KEELY, Royal R. (1901; 1907), Prof. Elec. Engrg., Nova Scotia Tech. College, Halifax, N. S.

KING, Roy Stevenson (1904; 1910), 252 Jefferson St., Valparaiso, Ind.

KUNZE, Edward J. (1907), Asst. Prof. Mech. Engrg., Michigan Agri. College, East Lansing, Mich.

LANGLOTZ, Robert (1894; Associate, 1904), 78 Delevan St., and *for mail*, 396 Third St., Brooklyn, N. Y.

LARNER, Chester W. (Associate, 1907), Hyd. Engr., Wellman-Seaver-Morgan Co., Cleveland, O.

LAUER, Conrad N. (Associate, 1903), Mgr., Dodge, Day & Zimmermann, Wash. Bldg., 608 Chestnut St., and 2217 Estaugh St., Philadelphia, Pa.

LEBRECHT, A. (1910), Cons. Engr., Hefnersplatz 3, and Bayrentherstrasse 22, Nuremberg, Germany.

LEEPER, Ralph W. (Junior, 1908), Genl. Elec. Co., Provident Bank Bldg., Cincinnati, O.

LEVIN, Arvid Michael (1894), Mech. Engr., 864 Monadnock Bldg., Chicago, Ill.

LINDSAY, W. Edward (1895), Cons. Engr., 1 S. 20th St., and *for mail*, P. O. Box 196, Birmingham Ala.

MILLER, Frank Louis (Associate, 1907), Erecting Engr., Southwark Fdy. & Mch. Co., Philadelphia, Pa.

MILNE, James (1907), Cons. Engr., Windsor, Ont., Canada.

PEDDLE, John Bailey (1909), Prof. Mch. Design, Rose Poly. Inst., and *for mail*, 2117 N. 10th St., Terre Haute, Ind.

PITKIN, Arthur Frederick (Junior, 1907), Asst. Mech. Supt., Genl. Elec. Co., Schenectady, N. Y.

RAY, David H. (Associate, 1904), Instr., College of the City of N. Y., 140th St. and Convent Ave., and *for mail*, 329 Edgecomb Ave., New York, N. Y.

REED, Edgar Howard (Junior, 1905), Supt., Reed & Prince Mfg. Co., Duncan Ave., and *for mail*, Wildwood Ave., Worcester, Mass.

ROSS, Charles Everett (1900), Cons. Engr., 52 Beaver St., and *for mail*, Hotel Chelsea, W. 23d St., New York, N. Y.

SHALLENBERGER, Louis R. (1895; 1902), Civ. and Mech. Engr., Paonia, Colo.

STREET, Clement F. (1893), with firm of Clement F. Street, Locomotive Stokers, 166 Washington Ave., and *for mail*, P. O. Box 192, Schenectady, N. Y.

STRUCKMANN, Holger (1909), Ch. Engr. and Genl. Mgr., Iola Portland Cement Co., Iola, Kan.

THOMAS, James Winthrop (1905), Mech. Engr., New England Engrg. Co., 50 Church St., New York, N. Y.

TRUELL, Karl O. (Associate, 1906), Colloseus Cement Co., 26 Beaver St., New York, N. Y.

TUTTLE, Irving E. (Junior, 1909), Mech. Draftsman, Westinghouse, Church, Kerr & Co., New York, and *for mail*, 475 Washington Ave., Brooklyn, N. Y.

WEGG, David S., Jr. (Junior, 1909), Telluride Inst., Olmstead, Provo, Utah, and *for mail* (temporarily), 16 E. Ontario St., Chicago, Ill.

WILDER, Clifton W. (1907), Elec. Engr., Pub. Service Com., 154 Nassau St., New York, N. Y.

WOOLSON, Ira H. (1896), Cons. Engr., Natl. Bd. of Fire Underwriters, 135 William St., New York, N. Y.

ZIMMERMANN, John Edward (Junior, 1907), Member of Firm, Dodge, Day & Zimmermann, Engrs., Wash. Bldg., 608 Chestnut St., Philadelphia, and Willow Grove, Pa.

NEW MEMBERS

BLAKESLEE, Frank Authur (1910), Mech. Supt., Taquah Min. and Exploration Co., Tarkwa, Gold Coast Colony, West Africa.

FEKETE, Stephen (Junior, 1910), 29 Spring Garden St., Edgewood Sta., Providence, R. I.

LUNDGREN, Charles G. (1910), M. M., National Tube Co., and *for mail*, 1018 W. 8th St., Lorain, O.

DEATHS

CRUIKSHANK, Barton, June 27, 1910

GARBETT, Joseph, August 22, 1910

PORTER, Charles T., August 28, 1910

SPIES, Albert, August 16, 1910

GAS POWER SECTION

CHANGES OF ADDRESS

ADAMS, Edward T. (1909), Mem.Am.Soc.M.E.
AUE, Jos. E. (1908), Mem.Am.Soc.M.E.
MILLER, Frank Louis (1910), Mem.Am.Soc.M.E.

STUDENT BRANCHES

CHANGES OF ADDRESS

ALLEN, Chas. L. (Student, 1910), Waterford, New York.
BOHNSTENGEL, Walter (Student, 1909), Dodge City, Kansas.
COLBURN, B. V. (Student, 1910), 911 Marion Ave., Cincinnati, O.
COYLE, J. F. (Student, 1909), 353 Wright St., Kingston Sta., Wilkes Barre, Pa.
FEHR, Roy B. (Student, 1910), 618 Kelly St., Wilkinsburg, Pa.
HELLER, Alfred (Student, 1909), Lackawanna Club, Lackawanna, N. Y.
HILFORD, V. H. (Student, 1909), Box 488, Caney, Kansas.
HODGSON, J. H. S. (Student, 1910), 89 Douglas St., Hammond, Ind.
JACOBSEN, C. H. (Student, 1910), 432 64th Ave., West Allis, Wis.
JAHNKE, Chas. B. (Student, 1910), care Mr. Northrop, Church St., Beloit, Wis.
JANDA, Jas. F. (Student, 1909), Y. M. C. A. Bldg., Aurora, Ill.
KERR, George W. (Student, 1910), 618 Kelly St., Wilkinsburg, Pa.
KONSTANKEWICZ, M. J. (Student, 1910), 116 W. 5th Ave., Gary, Ind.
LEHMAN, Myron G. (Student, 1910), 228 Elmwood Ave., Buffalo, N. Y.
LEVY, M. S. (Student, 1909), care Mrs. E. Stoefler, Amer. Shoe Store, 2 Rue du Quatre, Paris, France.
LOHSE, A. W. (Student, 1909), 3249 S. Park Ave., Chicago, Ill.
MANSFIELD, W. M. (Student, 1909), 312 Ferdinand Ave., Detroit, Mich.
MATTHAI, A. D. (Student, 1910), 1 Monroe Pl., Brooklyn, N. Y.
MERRIAM, Frank E. (Student, 1910), 815 South Ave., Wilkinsburg, Pa.
MOBLEY, Roy H. (Student, 1910), 1933 Market St., Philadelphia, Pa.
MONTGOMERY, Stafford (Student, 1910), Strafford, Chester Co., Pa.
NIXDORFF, S. P. (Student, 1909), 773 State St., Schenectady, N. Y.
PHILLIPS, Ernest C. (Student, 1910), Box 712, Gary, Ind.
POLHEMUS, D. A. (Student, 1910), 1106 Carnegie Bldg., Pittsburg, Pa.
PRIESTER, HENRY (Student, 1909), 21 Essex St., Hackensack, N. J.
REYNOLDS, Herbert B. (Student, 1909), 106 Highland Pl., Ithaca, N. Y.
SCHLEGEL, Carl A. (Student, 1910), 1005 University Ave., S. E., Minneapolis, Minn.
SCHULTZ, A. F. (Student, 1910), 1118 Franklin Ave., Wilkinsburg, Pa.
SMITH, Temple C. (Student, 1910), 38 Fulton St., Newark, N. J.
STEED, Arthur (Student, 1910), 824 Rebecca St., Wilkinsburg, Pa.
TRANE, R. N. (Student, 1910), 106 Lincoln Ave., Milwaukee, Wis.
WERNICK, F. E. (Student, 1909), 1110 13th St., Moline, Ill.
WHEDBEE, Edgar (Student, 1909), 190 S. Oxford St., Brooklyn, N. Y.
WOODS, Arthur P. (Student, 1909), Turtle Creek, Pa.
YOUNG, D. A. (Student, 1909), Delagua, Colo.

COMING MEETINGS

OCTOBER—NOVEMBER

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the editor's hands by the 15th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

ALABAMA LIGHT AND TRACTION ASSOCIATION

November 21-23, annual convention, Anniston, Ala. Secy., Geo. S. Emery.

AMERICAN ELECTROCHEMICAL SOCIETY

October 13-15, annual meeting, Chicago, Ill. Secy., Jos. W. Richards, Lehigh University, Bethlehem, Pa.

AMERICAN INSTITUTE OF MINING ENGINEERS

October 21-November 15, special meeting, Canal Zone, S. A. Secy., Dr. Joseph Struthers, 29 W. 39th St., New York.

AMERICAN MINING CONGRESS

September 26-October 1, annual convention, Los Angeles, Cal. Managing director, Sidney Norman.

AMERICAN RAILWAY ASSOCIATION

November 16, semi-annual meeting, St. Louis, Mo. Secy., W. F. Allen, 24 Park Pl., New York.

AMERICAN SOCIETY OF CIVIL ENGINEERS

October 5, 19, 220 W. 57th St., New York. Papers: October 5, Tieton Canal, E. G. Hopson; October 19, The Reconstruction of the Passenger Terminals at Washington, D. C., W. F. Strouse. Secy., C. W. Hunt.

AMERICAN SOCIETY OF MUNICIPAL IMPROVEMENT

October 11-16, annual convention, Erie, Pa. Secy., Prescott Folwell, 239 W. 39th St., N. Y.

AMERICAN STREET AND INTERURBAN RAILWAY ASSOCIATION

October 10-14, annual convention, Atlantic City, N. J. Secy., H. C. Donecker, 29 W. 39th St., New York.

AMERICAN RAILWAY BRIDGE AND BUILDING ASSOCIATION

Oct. 18-20, annual convention, Fort Worth, Ind. Secy., C. A. Lichty, 215 Jackson Blvd., Chicago, Ill.

ILLUMINATING ENGINEERING SOCIETY

October 24, annual convention, Baltimore, Md. Secy., P. S. Millar, 29 W. 39th St., New York.

INTERNATIONAL CONGRESS OF REFRIGERATION

October 9-12, University of Vienna, Austria. Papers: Constructing and Testing Refrigeration Machinery and Insulating Materials, Application of Refrigeration to Food, Industrial Refrigeration, Railway and Steam-

ship Refrigeration, Legislation and Administration. Chairman of Committee on Transportation, F. W. Pilsbury, 1660 Monadnock Block, Chicago, Ill.

NATIONAL ASSOCIATION OF RAILWAY COMMISSIONERS

November 16-23, annual meeting, Washington, D. C. Reports: Powers, Duties and Work of State Railway Commissions, Legislation, Shippers' Claims on Common Carriers, Railway Service and Railway Accommodation, Construction and Operating Expenses of Electric Railways, Simplification of Railway Tariffs, Rates and Rate Making, Uniform Classification, Car Service and Demurrage, Railway Capitalization, Safety Appliances, Grade Crossings and Trespassing on Railroads, Railroad Taxes and Plans for Ascertaining the Fair Value of Railroad Property, Amendment of Act to Regulate Commerce, Delays Attendant upon Enforcing Orders of State Railway Commissions. Secy., Wm. H. Connolly, c/o Interstate Commerce Commission.

NATIONAL MUNICIPAL LEAGUE

November 14-18, annual meeting, Buffalo, N. Y. Secy., Clinton Rogers Woodruff, North American Bldg., Philadelphia, Pa.

RAILWAY SIGNAL ASSOCIATION

October 11-13, annual meeting, Richmond, Va. Secy., C. C. Rosenberg, Bethlehem, Pa.

MEETINGS IN THE ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time p.m.
October			
5	Wireless Institute	S. L. Williams	7.30
5	Empire State Gas and Electrical Association	C. H. B. Chapin	All day
6	Blue Room Engineering Society	W. D. Sprague	8.00
11	American Society of Mechanical Engineers	C. W. Rice	8.15
13	Illuminating Engineering Society	P. S. Millar	8.15
14	American Institute of Electrical Engineers	R. W. Pope	8.00
18	New York Telephone Society	T. H. Lawrence	8.00
19, 20, 21	American Gas Institute	A. B. Beadle	All day
21	New York Railroad Club	H. D. Vought	8.15
26	Municipal Engineers of New York	C. D. Pollock	8.15
November			
2	Wireless Institute	S. L. Williams	7.30
3	Blue Room Engineering Society	W. D. Sprague	8.00
8	American Society Mechanical Engineers	C. W. Rice	8.15
10	Illuminating Engineering Society	P. S. Millar	8.15
11	American Institute of Electrical Engineers	R. W. Pope	8.00
15	New York Telephone Society	T. H. Lawrence	8.00
17–18	Society of Naval Architects and Marine Engineers	W. J. Baxter	All day
18	New York Railroad Club	H. D. Vought	8.15
23	Municipal Engineers of New York	C. D. Pollock	8.15

CURRENT BOOKS

THE RAILWAY LIBRARY 1909. A Collection of Noteworthy Chapters, Addresses and Papers Relating to Railways, Mostly Published during the Year. Compiled and Edited by Slason Thompson. *Chicago, The Gunthorp-Warren Printing Co., 1910.* Cloth, 8vo, 403 pp.

Contents: Introduction; Pre-Railway Era in America, by F. A. Cleveland and F. W. Powell; First Annual Report of the Chief Engineer of the Pennsylvania Railroad Company, by J. Edgar Thompson; Railways and the Pacific Northwest, by James J. Hill; Southern Railways and their Needs, by John F. Wallace; Problems confronting American Railways, by Daniel Willard; The Railroad Situation of To-day, by Frank Trumbull; Transportation Charge and Prices, by Logan G. McPherson; The Freight Rate Primer, issued by the N. Y. C. & H. R.R.R. Co.; Progressive Safety in Railway Operation, by A. H. Smith; Railway Mail Pay, by Julius Kruttschnitt; The Diminished Purchasing Power of Railway Earnings, by C. C. McCain; The Railroads and Public Approval, by Edward P. Ripley; Railroads and the Public, by John C. Spooner; Railroad Problems of Today, by J. B. Thayer; The Relation of the Railroads to the State, by W. M. Acworth; Railway Nationalization, by Sir George S. Gibb; Concerning Advances in Railway Rates, by the Senate Committee on Interstate Commerce, 1909; Statistics of American Railways for 1909, by Slason Thompson.

HIGH-SPEED STEEL. The Development, Nature, Treatment and Use of High-Speed Steels, together with some Suggestions as to the Problems involved in their Use. By O. M. Becker. *New York, McGraw-Hill Book Co., 1910.* Cloth, 8vo, 360 pp., Price, $4 net.

Contents: Introduction; Section I. The Development and Nature of High-Speed Steels; Section II. Making the Steels and the Tools; Section III. The Tool at Work; Section IV. The New Requirements; Section V. Problems Involved in the Use of High-Speed Steels; Section VI. Appendices.

FACTORY ORGANIZATION AND ADMINISTRATION. By Hugo Diemer. *New York, McGraw-Hill Book Co., 1910.* Cloth, 8vo, 317 pp., Price, $3 net.

Contents: Industrial Engineering; The Economic Theory of Factory Location; The Planning of Factory Buildings and the Influence of Design on their Productive Capacity; Staff and Departmental Organization;Executive Control in the Factory; Departmental Reports; The General Office; The Order Department; Bills of Material; The Drafting Department; The Pattern Department; Stores and Stock Departments; The Production Department; Foundry Systems; The Machine Shop and Tool Department; Shipping and Receiving Departments; Time Taking; Cost Department; Aids in Taking Inventory; Inspection Methods in Modern Machine Shops; Employment of Labor and Labor Problems; Wage Systems; The Fixing of Piece Work Rates; Principles Underlying Good Management; A Bibliography of Works Management.

COMPRESSED AIR. Theory and Computations. By Elmo G. Harris. *New York, McGraw-Hill Book Co., 1910.* Cloth, 8vo, 123 pp., Price $1.50.

Contents: Formulas for Work; Measurement of Air; Friction in Air Pipes; Hydraulic and Centrifugal Air Compressors; Special Applications of Compressed Air; The Air-Lift Pump; Examples and Exercises; Plates and Tables; Appendices.

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Terms expire at Annual Meeting of 1911

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Terms expire at Annual Meeting of 1912

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FRANCIS H. STILLMAN (2)
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PUBLICATION

D. S. JACOBUS (1), *Chairman*
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NOTE—Numbers in parentheses indicate number of years the member has yet to serve

SPECIAL COMMITTEES

1910

On a Standard Tonnage Basis for Refrigeration

D. S. Jacobus — G. T. Voorhees
A. P. Trautwein — Philip De C. Ball
E. F. Miller

On Society History

John E. Sweet — H H. Suplee
Chas. Wallace Hunt

On Constitution and By-Laws

Chas. Wallace Hunt, *Chairman* — F. R. Hutton
G. M. Basford — D. S. Jacobus
Jesse M. Smith

On Conservation of Natural Resources

Geo. F. Swain, *Chairman* — L. D. Burlingame
Charles Whiting Baker — M. L. Holman
Calvin W. Rice

On Identification of Power House Piping

H. G. Stott, *Chairman* — I. E. Moultrop
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On International Standards for Pipe Threads

E. M. Herr, *Chairman* — Geo. M. Bond
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On Standards for Involute Gears

Wilfred Lewis, *Chairman* — E. R. Fellows
Hugo Bilgram — C. R. Gabriel
Gaetano Lanza

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W. H. V. Rosing

On Power Tests

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Edward T. Adams — William Kent — Arthur West
George H. Barrus — Charles E. Lucke — Albert C. Wood

On Standardization of Flanges

A. M. Mattice — J. P. Sparrow
Wm. Schwanhausser — H. G. Stott

On Student Branches

F. R. Hutton, Honorary Secretary

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1910

On John Fritz Medal

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CHAS. WALLACE HUNT (3)
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On Board of Trustees United Engineering Societies Building

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On Library Conference Committee

J. W. LIEB, JR., CHAIRMAN OF THE LIBRARY COMMITTEE, AM.SOC.M.E.

On National Fire Protection Association

JOHN R. FREEMAN
IRA H. WOOLSON

On Joint Committee on Engineering Education

ALEX. C. HUMPHREYS
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On Advisory Board National Conservation Commission

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On Council of American Association for the Advancement of Science

ALEX. C. HUMPHREYS
FRED J. MILLER

NOTE—Numbers in parentheses indicate number of years the member has yet to serve.

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1910

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SECRETARY
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CHAS. E. LUCKE (3)
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Meetings of the Society in Boston

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EDWARD F. MILLER
I. E. MOULTROP, *Secretary*
J. H. LIBBEY

Meetings of the Society in St. Louis

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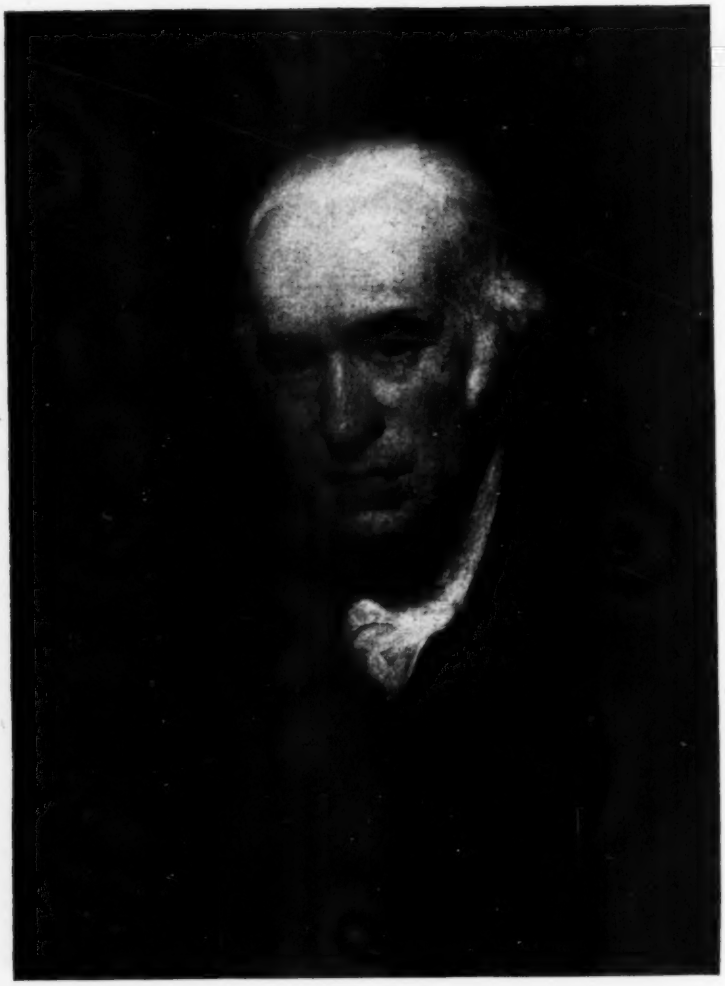
INSTITUTION	BRANCH AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	CORRESPONDING SECRETARY
	1908			
Stevens Inst. of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter	C. C. Allen	C. F. Hirschfeld
	1909			
Armour Inst. of Tech., Chicago, Ill.	March 9	G. F. Gebhardt	F. E. Wernick	W. E. Thomas
Leland Stanford Jr. University, Palo Alto, Cal.	March 9	W. F. Durand	A. F. Meston	J. B. Bubb
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianella
State Agri. College, Corvallis, Ore.	March 9	Thos. M. Gardner	C. L. Knopf	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	H. A. Houston	J. M. Barr
University of Kansas, Lawrence, Kan.	March 9	P. F. Walker	C. E. Johnson	C. A. Swiggett
New York Univ., New York	November 9	C. E. Houghton	Harry Anderson	Andrew Hamilton
Univ. of Illinois, Urbana, Ill.	November 9	W. F. M. Goss	B. L. Keown	C. S. Huntington
Penna. State College, State College, Pa.	November 9	J. P. Jackson	G. B. Wharen	G. W. Jacobs
Columbia University, New York.	November 9	Chas. E. Lucke	F. R. Davis	H. B. Jenkins
Mass. Inst. of Tech., Boston, Mass.	November 9	Gaetano Lanza	Morrill Mackenzie	Foster Russell
Univ. of Cincinnati, Cincinnati, O.	November 9	J. T. Faig	H. B. Cook	C. J. Malone
Univ. of Wisconsin, Madison, Wis.	November 9	C. C. Thomas	John S. Langwill	Karl L. Kraats
Univ. of Missouri, Columbia, Mo.	December 7	H. Wade Hibbard	R. V. Aycock	Osmer Edgar
Univ. of Nebraska, Lincoln, Neb.	December 7	C. R. Richards	W. J. Wohlenberg	W. H. Burleigh
	1910			
Univ. of Maine, Orono, Me.	February 8	Arthur C. Jewett	H. N. Danforth	A. H. Blaisdell
Univ. of Arkansas, Fayetteville, Ark.	April 12	B. N. Wilson	C. B. Boles	W. Q. Williams



JAMES WATT

PORTRAIT BY SIR WILLIAM BEECHY